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THE SCIENCE OF FLAMES  
AND FURNACES



# THE SCIENCE OF FLAMES AND FURNACES

*by*

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## PREFACE

Furnace research has long been the Cinderella of the industrial research organisations, partly because the use of furnaces came long before we knew anything of the physics or chemistry of the heating processes and partly because it is exceptionally difficult to isolate the different processes so that they can be studied separately in the laboratory. The study of combustion has been stimulated by the problems of the internal combustion engine to a very much greater extent than by those of the furnace, and in heat transfer forced and natural convection are very much better understood than luminous radiation, which is frequently the predominant mechanism of heat transfer in high-temperature furnace heating. Turbulence theory and the flow of fluids are very much more extensively studied in relation to aircraft design than they are in relation to furnace firing, although the value of the material handled annually in furnaces is many times greater than that of the aircraft industry. It is hoped that the present book may play a small part in helping to redress the balance by making such scientific knowledge as is already available more accessible to furnace designers and operators, and by bringing into focus the gaps in our scientific knowledge in such a way as to interest the more fundamental research workers in furnace problems.

The work of writing this book has been carried out while the author was on the staffs of the British Coal Utilisation Research Association and, later, the British Iron and Steel Research Association, and his grateful thanks are due to Mr. J. G. Bennett, first Director of B.C.U.R.A., who aroused his interest in these problems, and to Professor D. T. A. Townend, the present Director of B.C.U.R.A., and Sir Charles Goodeve, the Director of B.I.S.R.A., for many facilities and much help in the preparation of the manuscript. Dr. D. H. Bangham has also given a great deal of time and thought to the work, and many colleagues in B.C.U.R.A. and B.I.S.R.A. have greatly assisted. In particular, Mr. R. Mayorcas assisted the work on Chapter 1, Mr. J. H. E. Jeffes on Chapter 3, Mr. H. J. L. Herne on Chapter 4, Mr. M. P. Newby on Chapter 5, and Dr. J. H. Chesters and Dr. W. A. Archibald on Chapter 6. Miss V. I. Lewis and Miss H. Brown have assisted with illustrations and references. Mrs. M. A. Kirkham has given invaluable help and has been responsible for all the secretarial work.

*January 30th, 1950.*

M. W. T.

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## INTRODUCTION

Perhaps the most striking changes in the chain of processes that lead from mineral and organic raw materials to the finished articles in shops are those which occur inside furnaces. All the metals, glass, ceramics, cement, and even the building brick, depend essentially on processes in which the materials are heated through a cycle and either melted or undergo some physical or chemical change at high temperatures. A number of valuable books<sup>1.1</sup> deal with the design of industrial furnaces, but the present monograph attempts to approach the subject from a rather different standpoint in the hope of assisting to raise the industrial furnace from the status of Cinderella before the arrival of the Fairy Godmother. The furnace of the future will be a precision tool of the highest accuracy and reliability controlled by instruments as good as those in an aircraft and with every process within it as thoroughly studied as the circuit of a wireless set.

Some particular features of the mode of handling the problems adopted in the monograph are the following :

(i) It is intended here to stress the gap between such fundamental sciences as Physics, Chemistry and Physical Chemistry and industrial furnace design and use. For example, it is felt that a greater use of the science of thermodynamics in furnace design can lead to a more accurate and clearer way of thinking about such problems as those of preheat, use of waste-heat, recirculation of flue gases and different qualities of fuel. Consequently, it is hoped in this monograph to give a digest of such of the more fundamental work on these sciences as is likely to have immediate or ultimate application to furnace design.

(ii) The aspect of "Diagnosis" is particularly stressed. Diagnosis of the ailments of existing furnaces will always be an important branch of the scientific approach to the subject, since the calculations on which the furnace is designed must always involve many

- <sup>1.1</sup> TRINKS : *Industrial Furnaces*, vols. 1 and 2 (Wilson, New York).  
ETHERINGTON, H. : *Modern Furnace Technology* (Griffin, London).  
HERMANSON, A. : *Industrial Furnaces Technique* (Benn, London).  
MAWHINNEY, M. H. : *Practical Industrial Furnace Design* (Wiley, New York).  
GRISWOLD, JOHN : *Fuel, Combustion and Furnaces* (McGraw Hill, 1946).  
HASLAM AND RUSSELL, *Fuels and Their Combustion*.



## INTRODUCTION

uncertain assumptions, e.g. those connected with the behaviour of the very complicated solid, liquid and gaseous materials handled. Hence design must always remain an art as well as a science, and it follows that there must always be a greater or lesser divergence between the operating results of the furnace and the calculations. Diagnosis can be made either by the use of the experienced eye behind a coloured glass or by the use of scientific instruments as probes. Various examples of the latter method are discussed in Chapters 3, 4 and 5 of this monograph.

(iii) Finally, the attempt is made to present the problems of furnace heating in such a way as to be of interest to the academic scientist as well as to the engineers in charge of the design and operation of furnaces. Such engineering products as steel bridges, electric transformers, wireless sets, steam turbines and aerofoils have advanced considerably from the empirical stage as a result of the interest of the academic scientists. It is accordingly felt that if the scientific questions to which answers are necessary can be stated with sufficient accuracy, it may make a small contribution towards the improvements which are so clearly necessary.

In order to fulfil these three aims it is considered necessary to limit the scope of the monograph very severely in certain directions. Thus no mention will be made of furnaces other than those heated by flames, i.e. all types of electrical furnace will be entirely omitted.<sup>1.2</sup> Also, problems relating more to the material being heated than to the furnace itself will be omitted: for example, methods of handling the charge will be considered only in so far as they set limitations on the design of the furnace itself.

- <sup>1.2</sup> The following books deal with the design of industrial electric furnaces :  
ROBIETTE, A. G. : *Electric Melting Practice* (Griffin, London).  
TAUSSIG, R. : *Elektrische Schmelofen* (Julius Springer, Wien, 1935).  
PASCHKIS, V. : *Industrial Electric Furnaces*, vols. 1 and 2 (Interscience Publishers, New York).



## 1. THE FUNCTION OF A FURNACE

### 1.1. What is a Furnace?

It is not possible to make any rigid logical definition of what is meant by a furnace, any more than it is possible to make such a definition for a simple thing like a chair. As a practical working definition, however, it is proposed to restrict the term to an industrial appliance constructed to heat a material through a cycle involving temperatures exceeding  $400^{\circ}\text{C}$ . This temperature has been chosen in order to exclude a large number of industrial processes in which steam is used as a medium for transferring heat.

The designer or purchaser of a furnace has a threefold aim. First and foremost, it is necessary to obtain a particular processed product easily and reliably. For this purpose the furnace must have very good control of temperature, of temperature cycle, and of solid, gaseous and liquid substances which are allowed to come in contact with the material to be processed.

Secondly, the furnace must require a minimum amount of fuel and other auxiliary materials. Economy of fuel depends, primarily, on two things: firstly the *direct* use of as much as possible of that part of the combustion energy which is above the temperature of the coolest part of the charge in the furnace, by insulation, good heat transfer and counterflow heat exchange between combustion gases and charge. Secondly it depends on the *indirect* use of as much as possible of the heat below this temperature, usually by preheaters or waste-heat boilers or even, as recently suggested, to operate an oxygen separator to provide enriched combustion air.

The third aim of the furnace designer is to obtain the results with a minimum of capital and maintenance costs. The achievement of this aim is primarily dependent on the output that can be obtained with a given size of furnace, on the simplicity of design\* and on the availability of satisfactory construction materials.

\* In industries where the economic situation demands an output greater than that for which the furnace was laid down (e.g. the steel industry during a war period) this third aim is usually regarded as being of greater importance than the second.



## 1.1 THE SCIENCE OF FLAMES AND FURNACES

In order to discuss what are the essential parts of a furnace it is useful to make use of an analogy which has already been suggested by the use of the word "diagnosis" in the Introduction. This analogy is between the furnace and the living creature. Thus the first essential of a furnace is a "skin," or boundary, separating the region maintained at a desired temperature and the surrounding atmosphere. In the case of a furnace this skin is clearly the refractory walls : their function is to allow a minimum exchange of heat and gases with the outside except at the desired points. Just as in the case of an animal the breath and food which go into the body have to be heated to the internal temperature on their way in, so the materials entering the furnace have to be brought to the prevailing temperature on their way in. These materials may be broadly classified as the raw materials or charge, the fuel or energy source and the air for combustion. Thus from this analogy we can see that the essential parts of the furnace are :

- (1) the wall structure that differentiates it from its surroundings ;
- (2) the apparatus by which the three main materials enter it and the products leave ; and
- (3) the mechanism which maintains the desired conditions and allows them to be controlled by the operator.

## 1.2. The Furnace-using Industries

In order to understand the factors which constitute the success of a furnace it is necessary to examine the industries in which it is used. The purpose of this section is to give a brief account of the role of some important industrial furnaces in their industries,\* together with a critique of the effectiveness with which they perform their role. It will appear that imperfection may be due either (*a*) to imperfections in the materials available for construction ; (*b*) imperfections in the basic principle of the system, i.e. failure to be along lines which can ultimately give exactly what is wanted ; and (*c*) imperfections in design. The present chapter criticises them mainly from the second point of view ; Chapters 2, 3, 4 and 5 discuss the knowledge which may lead to perfection in design ; and Chapter 6 discusses the possibility of improving the materials of construction.

\* For a classification of furnaces, as furnaces rather than by industries, see Section 1.4 below.



**1.2.1. Iron.** The blast furnace,<sup>1,3</sup> Fig. (1)1, is the largest industrial furnace in use, and outputs of up to 1400 tons of pig iron per 24 hours are obtained from a single furnace. The blast furnace is perhaps the most complicated furnace system, as it combines the operations of reducing the iron oxide, melting the iron and the slag and separating the latter by gravity. It consists essentially of a

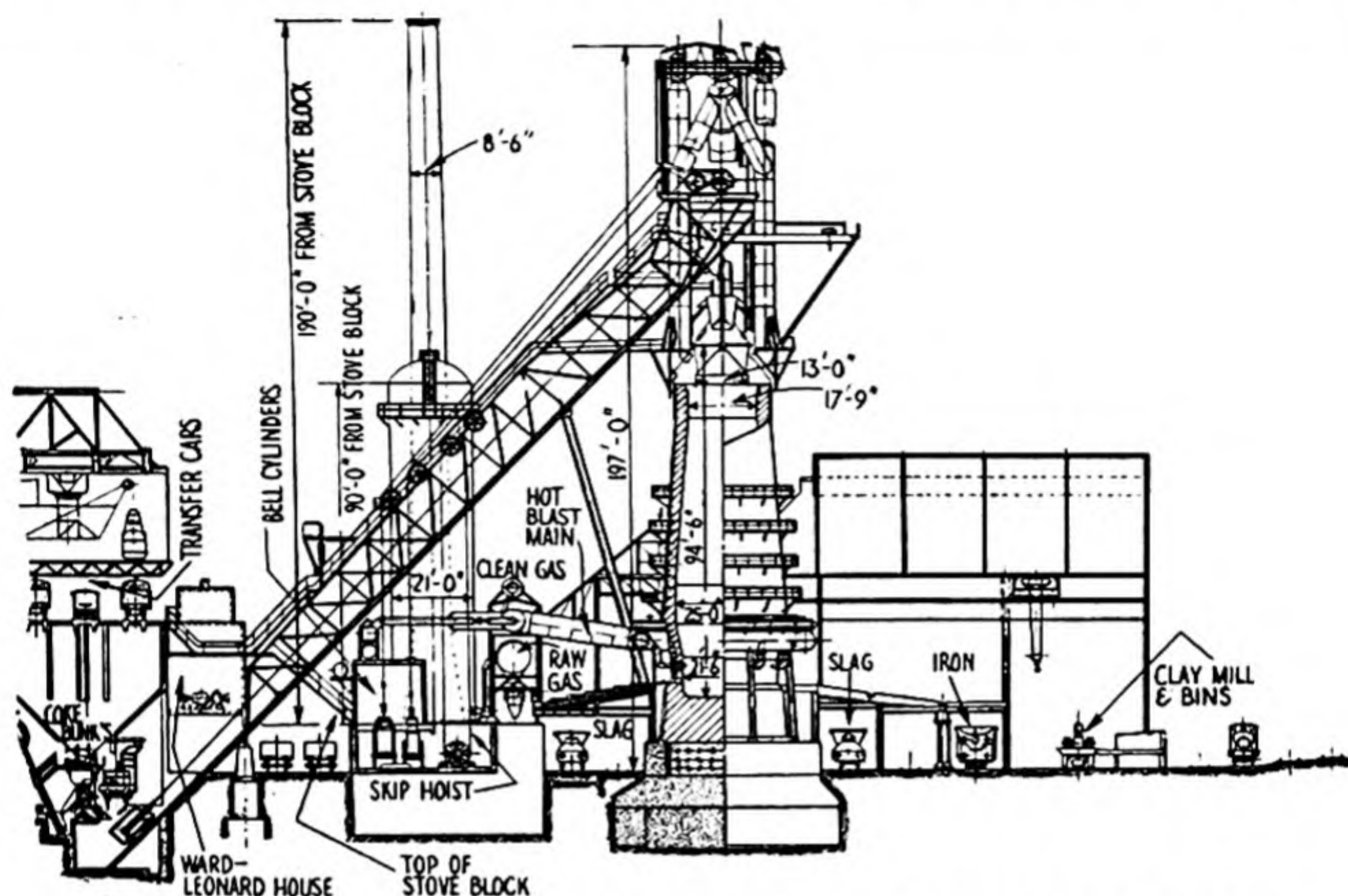


Fig. (1)1. Cross-section through Blast Furnace Plant at Margam Works.  
(Courtesy of Iron and Steel Institute.)

vertical brick-lined steel shaft, with a hearth or crucible at the bottom to hold molten metal which has a diameter between 8 and 27 ft. This shaft tapers outwards, going down from the top to reduce bridging of the materials, then has a short parallel portion,

- <sup>1,3</sup> JOSEPH, T. L. : U.S. Bureau of Mines Information Circular 6779.  
For a more recent account of blast furnace development see :  
HAVEN, W. A. : "The Manufacture of Pig-Iron in America," *J.I.S.I.*, **141**, No. 1, p. 605P (1940).  
ELLIOT, G. D. : "Iron-Making at Appleby-Frodingham," *I.S.I. Special Report*, No. 30.  
BROWN, W. R. : "The Blast Furnace of Today. A Review of Current Furnace Engineering" (Part I), *J.I.S.I.*, **155**, I., p. 107 (Jan. 1947).  
SCOTT-MAXWELL, I. S. : "A Commentary on Current Furnace Engineering" (Part II), *J.I.S.I.*, **155**, p. 115.  
HENDRY, D. C. : "Engineering Problems in the Preparation of Ores for Blast Furnaces," *J.I.S.I.*, **155**, p. 121 (Jan. 1947).  
COLCLOUGH, T. P., and SCOTT-MAXWELL, I. S. : "Trends in Pig Iron Manufacture," *J.I.S.I.*, **159**, Part II, p. 186 (June 1948).



### 1.2.1 THE SCIENCE OF FLAMES AND FURNACES

and tapers inwards rather rapidly to the hearth, which is usually slightly smaller than the top of the shaft. Coke, iron ore and limestone (as a flux for the impurities in the ore) are fed in at the top through a gas seal (double bell), beneath which is situated the offtake for the blast furnace gas. Air, supplied by steam-driven turbo-blowers, is preheated by passing it through first one and then another of three brick-filled chambers (Cowper stoves), which are heated in turn by burning some of the blast-furnace gas in them, the air being then blown with high velocity into the furnace through a ring of water-cooled copper tuyeres situated just above the hearth. The high velocity is intended to force the air to distribute itself right across the shaft,<sup>1,4</sup> but in practice active combustion only takes place in a region extending inwards about 40 inches from the tuyere. Near the tuyeres the air consumes coke to form  $\text{CO}_2$  and then is reduced to CO by more coke. This CO reduces iron ore in the next part of the shaft, while in the topmost part it mainly heats the column. The final gases, which contain 20–30% CO, leave the top of the shaft below the double sealing bell and pass to various uses (heating the Cowper stoves, coke ovens, furnaces, boilers and in some cases supplying electrical current from gas engines). Below the tuyeres the molten iron, saturated with carbon, and the molten slag collect. They are separated by gravity and can be tapped off through two separate tapholes one above the other. Since the fuel acts as a consumable support for the burden, allowing it to flow down freely, maintaining permeability in the bed and preventing adhesion, in addition to its function as a source of heat and chemical reduction, it is very difficult to devise a scheme which will eliminate the need for high-grade coke free from excessive fines and at the same time give the tremendous output of a modern blast furnace. Moreover, the essentially counterflow character of the blast furnace, although not quite ideal from the thermodynamic standpoint (the ideal process would bring the fuel into admixture with the rest of the burden at some intermediate point between the top of the stack and tuyeres), is very nearly so, especially if the blast-furnace gas is put to good use in the plant.

A good deal is known about the chemical processes in the blast furnace, the reduction and oxidation taking place between the gases, the coke and the iron ore. It is, however, a fundamental limitation of this process that the gases leaving the top of the shaft must contain a larger percentage of CO than  $\text{CO}_2$  and hence that a large

<sup>1,4</sup> KINNEY, S. P. : U.S. Bureau of Mines, 1929, *Tech. Paper* 442.



proportion (35-50%) of the energy of the coke cannot be directly used in the furnace, while high-grade coke is a very expensive source of low calorific value gas. As the efficiency of the system is gradually raised the  $\text{CO}/\text{CO}_2$  ratio falls and the blast furnace gas becomes less and less valuable as a fuel. This indicates that the use of slight oxygen enrichment of the blast<sup>1.5</sup> may be valuable up to a point, as it will enable the necessary temperatures to be reached with a smaller amount of chemical energy carried away in the gas. Too oxygen-rich a blast will, however, give insufficient heat carried up the stack to heat the downcoming materials.

Other basic imperfections of the blast furnace, apart from its need for an expensive prepared fuel much of which is turned into low-grade gas, are that the iron is necessarily saturated with carbon and that all the slag from the impurities in the iron ore and the coke ash has to be melted. Since, however, it is essential that the iron be melted both to bring it into a convenient form for further use and to reduce the sulphur content, it is difficult here again to devise alternative schemes which overcome these disadvantages.

Recent research work on the blast furnace<sup>1.6</sup> has therefore been concentrated mainly on the study of the flow of the gases up the shaft and the flow of the solid materials down the shaft,<sup>1.7</sup> the aim being to achieve uniformity in both cases. The use of carbon bricks to line the hearth and reduce the formation of a "bear," or solid block of iron extending deep into the foundations, is now being extended.<sup>1.8</sup>

The mechanism of the heat transfer in this type of furnace has not been the subject of much investigation, largely because heat transfer does not represent a bottleneck, the surface of contact

<sup>1.5</sup> THRING, M. W.: "Possibilities for the extended use of Oxygen in the British Iron and Steel Industry," *J.I.S.I.*, **156**, Pt. 2, p. 275 (1947).

<sup>1.6</sup> SAUNDERS, H. L.: Article compiled by T. Bishop, "Research in Iron Making Work in Progress in the British Iron Making Industry," *Iron and Coal Trades Review*, 31 Oct. 1947.

SAUNDERS, H. L., BUTLER, G. B., and TWEEDY, J. M.: "Investigations on an Experimental Blast Furnace," *J.I.S.I.*, **163**, Pt. 2, p. 173 (Oct. 1949).

FORTUNE, J. B.: "Progress in Blast Furnace Research," *Iron and Steel*, 1939.

<sup>1.7</sup> CHESTERS, J. H., HALLIDAY, I. M. D., and MACKENZIE, J.: "An Examination of Blast Furnace Scaffolds," *J.I.S.I.*, **159** Pt. 1, p. 23.

SAUNDERS, H. L., and WILD, R.: "Distribution of Materials in the Blast Furnace," *J.I.S.I.*, **163**, Pt. 1, p. 61 (1949).

<sup>1.8</sup> SINGER, F.: "Carbon Linings for Blast Furnaces," *Metals and Alloys*, **10**, p. 105 (1939).



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between gases and solids being very great (see, however,<sup>1.9</sup>, where a study of the heating up to beds of spheres by flowing gases is described). Numerous alternative systems for making pig iron from ore have been proposed in the attempt to overcome the defects discussed above, notably one based on the rotary kiln,<sup>1.10</sup> Fig. (1)2, and some have been tested on a commercial scale.<sup>1.11</sup> None have been able to compete on their own ground with the blast furnace, owing to the enormous output per unit of the latter, but the low-shaft electric smelting furnace is in commercial operation in Sweden and Switzerland,<sup>1.12</sup> where electricity is readily available from water power, while coke is expensive. Recently also experiments have been carried out in Germany with a low-shaft smelting

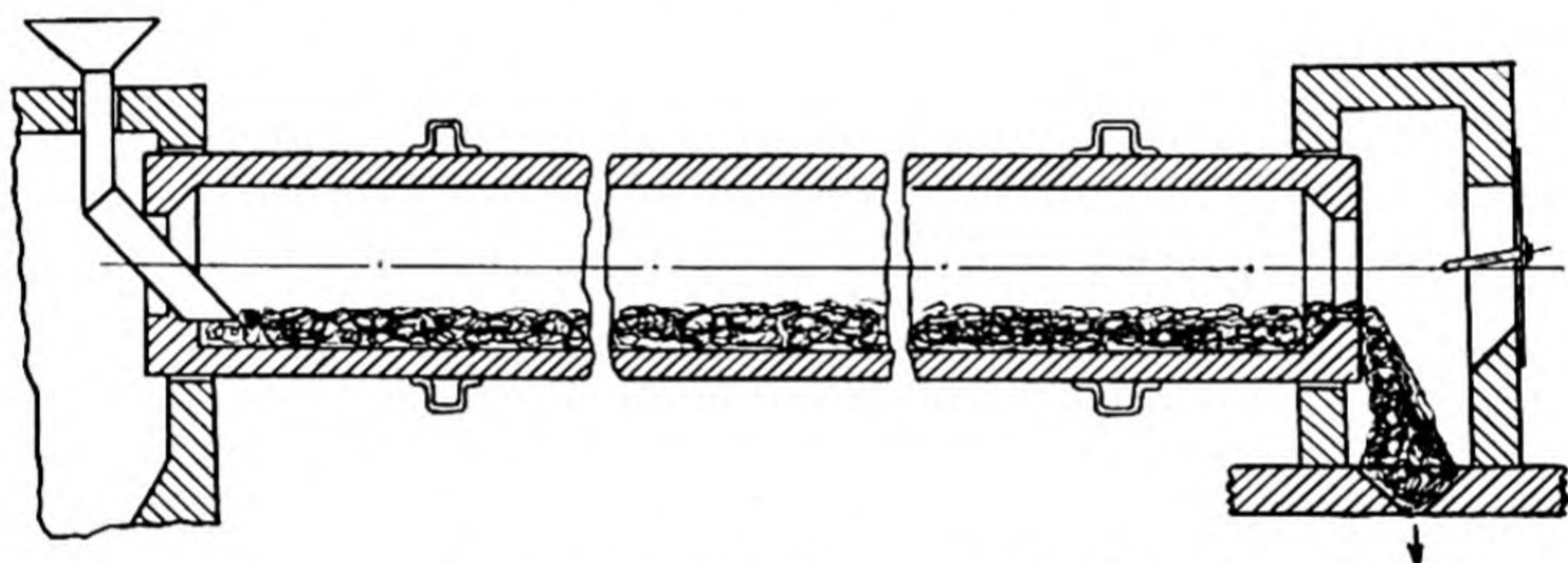


Fig. (1)2. Krupp-Renn Rotary Kiln for the Production of Iron Nodules.

furnace using a cold blast containing up to 100% oxygen, but more often oxygen in the range 60–80%. These showed that basic pig iron can be made in such a unit, but with considerably greater fuel consumption than with normal preheated air in a high-shaft furnace.

The apparatus is of interest, however, for the manufacture of certain alloys which are normally made electrically, such as high-carbon ferro-chrome and ferro-silicon-chrome.

<sup>1.9</sup> SAUNDERS and FORD : "Heat Transfer in the Flow of Gas through a Bed of Solid Particles," *J.I.S.I.*, **141**, p. 291 (1940/41).

<sup>1.10</sup> JOHANNSEN, F. J. : "The Krupp-Renn Process," *Stahl und Eisen*, p. 969 (20 Sept. 1934).

<sup>1.11</sup> A very full review of those alternative processes for the reduction of iron ore which are in successful operation particularly in Sweden, and of many unsuccessful ones, is given by E. C. Smith : "Evidence to Date on Iron Production by Methods Other than Coke Blast Furnace ; Direct Reduction," 1948 *Yearbook* of the American Iron and Steel Institute, p. 302.

<sup>1.12</sup> DURRER, R. : "Electric Smelting," *J.I.S.I.*, **156**, Pt. 2, p. 257 (June 1947).

The cupola furnace,<sup>1-13</sup> Fig. (1)3, may perhaps be regarded as a small-scale edition of the blast furnace except that the upper part of the shaft does not have to reduce iron oxide to any great extent, but has only to be the site of heat exchange between the hot combustion gases and the descending coke and pig iron. The walls are

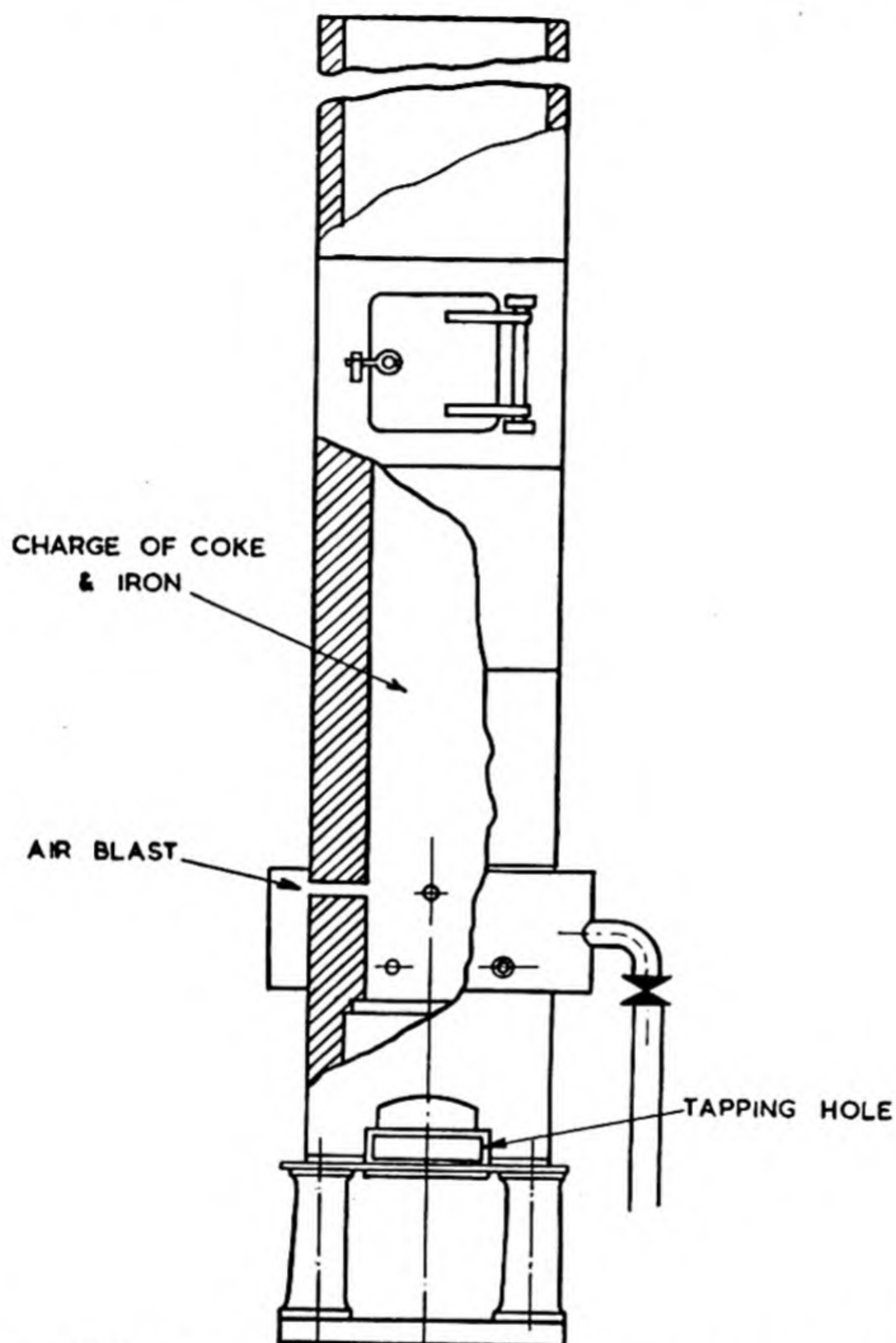


Fig. (1)3. Cupola Furnace for Iron Casting Manufacture.

substantially cylindrical, since there is not the problem of sticky ores which occurs in the blast furnace. Fuel economy in the cupola depends mainly on the success which is achieved in lowering the CO content of the gases leaving the top of the shaft by correct

<sup>1-13</sup> SCHMIDT, L. : *Der Bau u. Betrieb der Kupolofen* (von Wm. Knapp, Halle (Salle), 1933).



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feeding and choice of size of materials fed. In some cases, also, the air is preheated in recuperators or regenerators heated by the waste gases. This has a marked effect on fuel economy.\* The cupola can melt iron with a fuel consumption which is very low in comparison with furnaces in which the fuel and charge are separated, but it naturally is much more restricted as to the quality of the iron produced. The carbon content of the metal is lower than blast furnace iron, being usually 1-2%. The use of large cupolas to

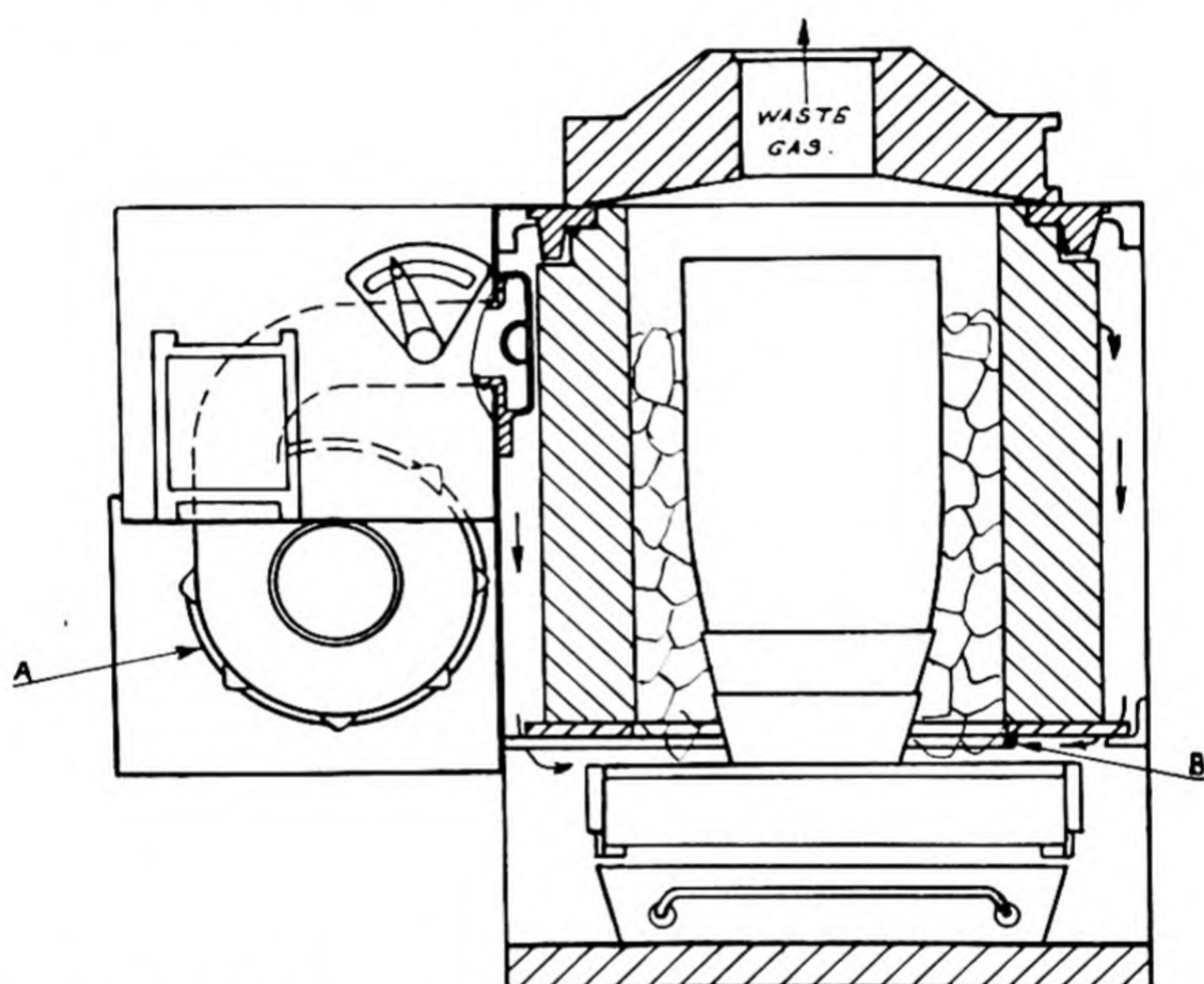


Fig. (1)4. Coke Fired Crucible Furnace for Small Iron or Steel Melts.  
(Courtesy of Morgan Crucible Company Limited.)

melt scrap steel instead of supplying this scrap cold to the open-hearth furnace is one possibility which is very promising for steel-works of the future. Other furnaces in use in the iron industry are the crucible furnace for small castings, Fig. (1)4, and the Stuerzelberg rotary furnace,<sup>1-14</sup> Fig. (1)5.

**1.2.2. Steel.** The great bulk of steel (85% of the total in this

\* See PIWOWARSKY, E.: "Über das Kupolofenschmelzen mit überhitzten Heisewind," *Die Gusserei*, **26**, p. 169 (1939). He shows that there is a decided advantage in fuel economy and iron quality in using pre-heat up to 700° C. Also LONGDEN, E.: "Hot-Blast Cupola Design," *Foundry Trade Journal*, **71**, pp. 387 and 421 (15 and 22 Aug., 1946).

<sup>1-14</sup> *Stahl und Eisen*, No. 1, 1937; *Iron Age*, **14**, p. 1 (1938).

country) is made in the open-hearth (or Siemens-Martin) furnace,<sup>1.15</sup> Fig. (1)6, which does not differ in physical principles from that developed in 1860–70 by Sir William Siemens, although the bath capacity has increased on the largest furnaces from the original 2 tons of Siemens' first furnace to the largest one now operating, of 550 tons ; while the heat-absorbing surface area of the largest ones is now over 1000 ft.<sup>2</sup> Engineering details also have altered very greatly alongside the growth in size, some marked alterations being the tilting

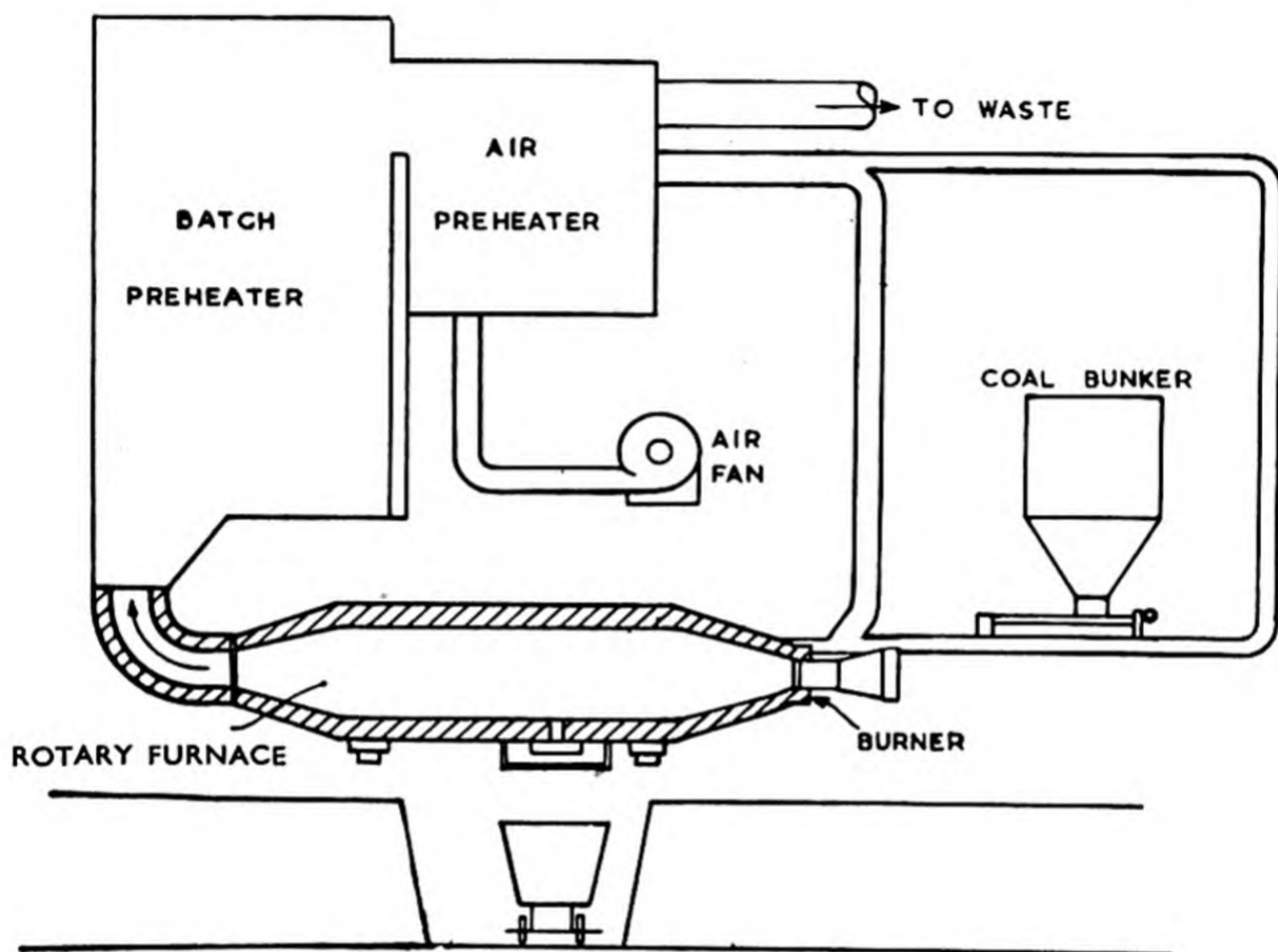


Fig. (1)5. Stuerzelberg Rotary Furnace for Iron Melting.

furnace, plated checkers, straight-through reversing valves and the introduction of instrumentation and automatic control (see 7.2.5).

Various steelmaking processes are in use in the open-hearth furnace, but in all of them a certain percentage of steel scrap is melted and a further quantity of pig iron is either melted or raised in temperature by heat transfer (mainly radiation) from a flame passing over the surface of a bath. During and after these melting processes reactions take place between flame, slag and steel which result in the carbon content of the molten metal being reduced to a

<sup>1.15</sup> Symposium on Steelmaking, *I.S.I. Special Report* No. 22, 1938.

LECKIE, A., "Recent Developments in Open Hearth Furnace Design and Operation," *Engineering*, p. 599 (2 Dec. 1949).



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desired figure, while other impurities, notably sulphur, phosphorus and silicon, are also removed to a greater or less degree. The actual steelmaking processes used in the open-hearth vary in the following respect. In the first place there are the "hot metal" process, in which the pig iron is supplied molten from a blast furnace via a mixer (which may be *inactive*, in which case it is simply a tilting holding vessel of 1000 tons or more capacity ; or *active*, in which case it is really a tilting open hearth with a deep bath) ; and the "cold metal" process, in which the pig is frozen ; and

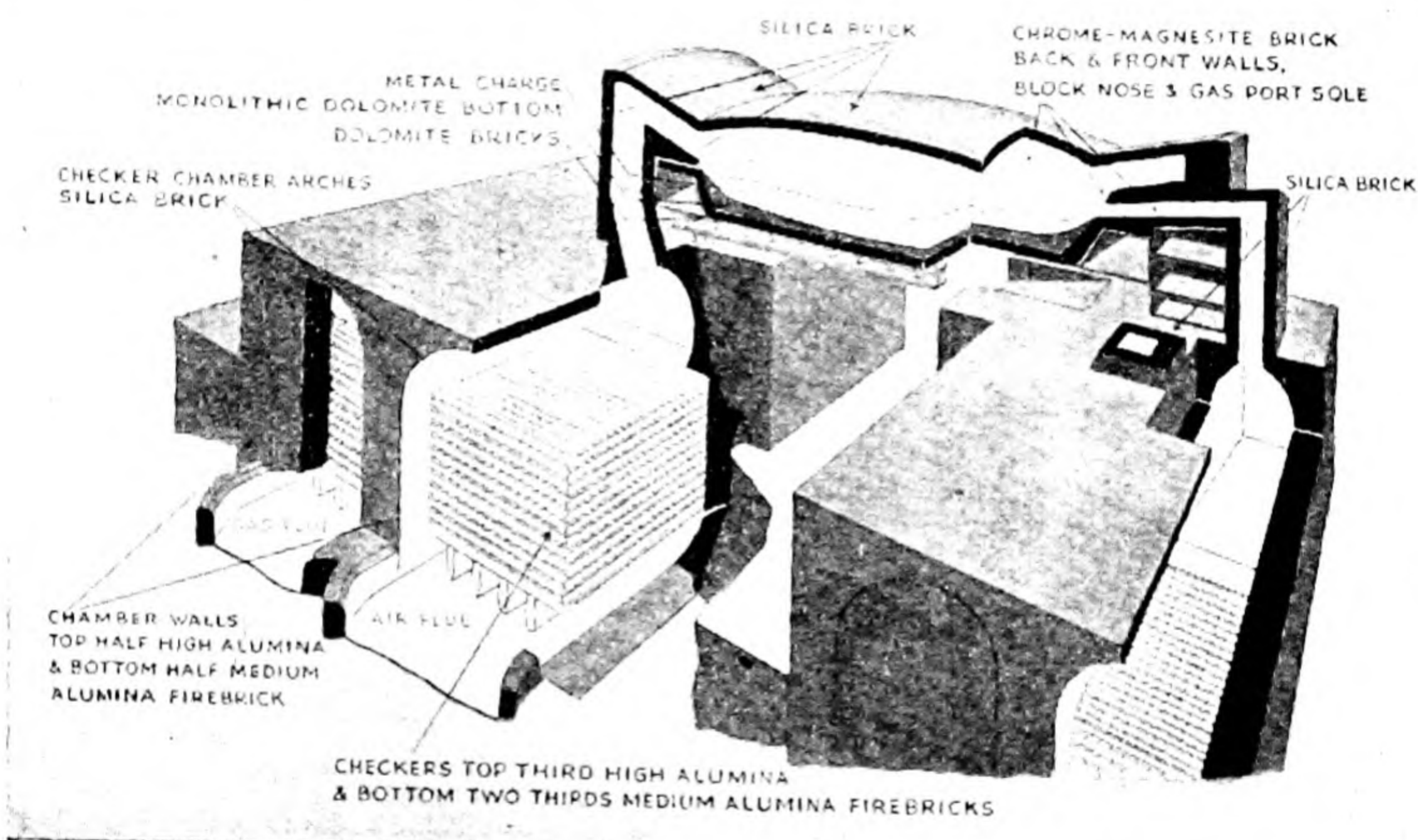


Fig. (1)6. Cold Pig Fixed Open-Hearth Steel Melting Furnace sectioned to show gas flow passages.

(Courtesy of The United Steel Companies, Limited.)

thirdly, a process much used in America, the "blown metal" or duplex process, in which the mixer metal is blown down to a carbon content of about 1% in a Bessemer converter before being fed to the open hearth. The relative economics of these three processes depend on the relative prices of scrap and pig, a low scrap price favouring the cold pig process. Other variations in the process arise according to whether an intermediate slag flush off is used, in the extent to which iron ore is fed to assist in the oxidation of carbon in the metal, and in the use of an acid (silica) or basic (magnesite or dolomite) hearth according to the quality of the steel to be made,



the former giving a slower, more expensive process but a steel with less oxide inclusions.

The physics of a heat transfer system with the source of heat above the sink of heat, in which the solid sinks in the liquid and in which a liquid slag layer forms, are of considerable interest.<sup>1.16</sup> As soon as the piled-up material is melted below the liquid surface the system bears a close relation to the well-known laboratory experiment in which water is boiled at the top of a test-tube while ice, weighted down with a wire gauze, remains unmelted at the bottom—in both cases heat has to be transferred the wrong way for natural convection and hence considerable temperature gradients arise.

The necessary flame temperature to transfer heat to a bath, when the metal may be at a temperature up to 1600° C. and with an appreciable temperature gradient in the slag, is of the order of 1700–1800° C. and even higher. This temperature is attained in all open-hearth steel-melting furnaces by means of regenerative preheaters for the air and also for the fuel when the latter is producer gas or mixed coke-oven and blast-furnace gas. Although counter-flow recuperative preheaters in which the waste gases leaving the furnace and the incoming air flow in opposite directions along either side of a system of more or less gas-tight ceramic or metal walls have been tried for the open-hearth furnace, they have not so far proved anywhere near an economic competitor of the original Siemens regenerator because the recuperator makes much more stringent requirements on the heat transfer surfaces, which must not crack nor build up with slag, and they cannot give such a high degree of preheat.

The regenerator was suggested to Siemens by a device for warming the air breathed by consumptive patients, consisting of a box filled with loose metal packing through which the patient breathed in and out. It consists of a pair of refractory boxes filled loosely with brickwork, through one of which the gases leaving the furnace pass and heat up the brickwork, while the incoming air passes in the opposite direction through the other and picks up heat from it. Every 10–15 minutes the valves connecting the atmosphere to the cold end of one box and the base of the stack to the cold end of the other are reversed. In this system even the inside of the passages leading from the regenerators to the furnace provides useful heat transfer capacity, and the regenerators can handle gases

<sup>1.16</sup> THRING, M. W.: "The Influence of Heat Transfer on Open Hearth Furnace Charging Rate," *J.I.S.I.*, 161, p. 212 (March 1949).



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at the full furnace temperature and give a preheat to the air of  $1000^{\circ}$ – $1200^{\circ}$  C. and even higher. It is perhaps unfortunate that such tremendous theoretical flame temperatures can be obtained in this way that the requisite furnace temperature is easily achieved, so that it has not been necessary to make the leak-proof open-hearth furnace which would lead to very considerable fuel savings. The chief disadvantages of the regenerative system are that it superimposes a short-period fluctuation on all the furnace variables in addition to the long-period one set by the fact that the steelmaking process is itself of the batch type, and that it restricts the design of the furnace ends, since they have to serve both as exits for waste gases as well as entrances for preheated air and fuel. Special points about the design of the open-hearth furnace arise from the fact that silica, the only material found in very large quantities in the pure state in nature which is suitable for making the crowns and other parts, melts at  $1720^{\circ}$  C. and cannot be used in practice with a hot-face temperature above  $1680^{\circ}$  C., so that there is a very small margin between the required and the safe maximum temperature. Hence the life of an open-hearth furnace crown is of the order of 10–20 weeks, there can be very little insulation, great attention has to be paid to directing the flame on to the charge, and in many furnaces some parts where the gases turn a sharp corner must be repaired every few days. Such limitations on design can only be removed by showing that higher quality, but rarer, refractories justify the initial high cost. The use of the all-basic furnace with a chrome-magnesite crown is now under extensive investigation.

The main function of the furnace is to melt the steel and slag\* and then to supply a suitable environment in which the reactions between the molten metal, the slag and oxygen in the combustion gases can take place to reduce the elements (C, Si, P and S) in the former to the desired quantities. Carbon and silicon are reduced primarily by iron oxide in the slag and by oxidation from the flame,<sup>1.17</sup> while the phosphorus and sulphur contents of the metal

\* This consists of silica and iron oxide (acid process) and silica iron and lime (basic process).

<sup>1.17</sup> For an analysis of the steelmaking reactions see :

SCHENCK, H. : *Physical Chemistry of Steelmaking* (B.I.S.R.A.).

MUDD, Seeley W. : "Basic Open-hearth Steelmaking," *A.M.I.E. Committee on Physical Chemistry of Steelmaking. Iron and Steel Division. New York, 1944.*

WASHBURN, F. M., and PHILBROOK, W. O. : *Iron Age*, p. 21 (22 Feb. 1940); p. 31 (29 Feb. 1940); p. 59 (7 March 1940).



are lowered in the basic furnace by reaction with slag containing a high proportion of  $\text{CaO}$ .

In spite of its great popularity, the open-hearth has certain disadvantages from the purely scientific standpoint. (i) Although the passage from cold raw materials to cold steel ingot or from molten pig iron to molten steel is exothermic, the open-hearth process uses  $2-5 \times 10^6$  C.H.U./ton of steel of fuel heat supply. While this figure could be appreciably reduced by improving the flame conditions (higher  $\text{CO}_2$  and luminosity), by using better refractories which permit insulation, and by changing the steelmaking to render easier the passage of heat into the bath, it will remain an inherent defect of this system that it needs fuel to carry out an exothermic process. This objection applies to all methods of firing used in the open-hearth furnace and is one of the principal points in favour of the Bessemer process. (ii) When coal is the fuel it has so far been necessary\* to convert it to producer gas, and thus the system has to include producers, mains carrying hot producer gas in which the tar content of the gases frequently condenses, reversing valves, four regenerators, slag pockets, ports, furnace and stack, so that it is extremely complex and costly. Even the oil-fired system remains fairly elaborate although there are only two checkers and the producer mains are replaced by a heated tank and lagged pipe lines.<sup>1.18</sup> Among the most important of other fuels are: (1) a mixture of blast-furnace and coke-oven gas of about 220 B.Th.U./ft.<sup>3</sup>, the mixture being preheated in separate checkers as with producer gas; and (2) cold coke-oven gas of about 500 B.Th.U. together with tar to increase the luminosity of the flame. The "integrated" steel works in which the only raw coal is that supplied to the coke ovens represents the best economic practice for using one or other of these fuels. Pulverised coal has been tried in the past, but abandoned owing to the effects of the coal ash, but recently interest in this possibility has to some extent revived. (iii) As mentioned above, to obtain a maximum heat transfer rate in melting a solid the heat should ideally be applied from below so as to supplement conduction in the material already melted by convection. The open-hearth does exactly the opposite; this is

\* See, however, SIEMENS, C. W.: *J.C.S.*, **21**, p. 279 (1868). This paper describes the "new form" Siemens furnace which consisted of a built-in gas producer using hot valves, and only the air was pre-heated in the regenerators.

<sup>1.18</sup> LEMON, F. A.: "The Production in Rotary Furnaces of Steel for Castings," *J.I.S.I.*, **156**, p. 53 (1942).



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probably not a serious heat transfer bottleneck when the slag blanket is thin, but in certain practices where it has to be thick, especially when it is necessary to make very low-sulphur steel from raw materials containing considerable sulphur and silicon, it can greatly slow down the process. On the other hand, it seems almost essential to put the heat in through the liquid surface with steel because otherwise it is necessary to have a refractory container with the temperature of molten steel on the inside and a considerably higher one on the outside.

The principal rival to the open-hearth furnace for steelmaking is the Bessemer converter, Fig. (1)7, in which air is blown upwards

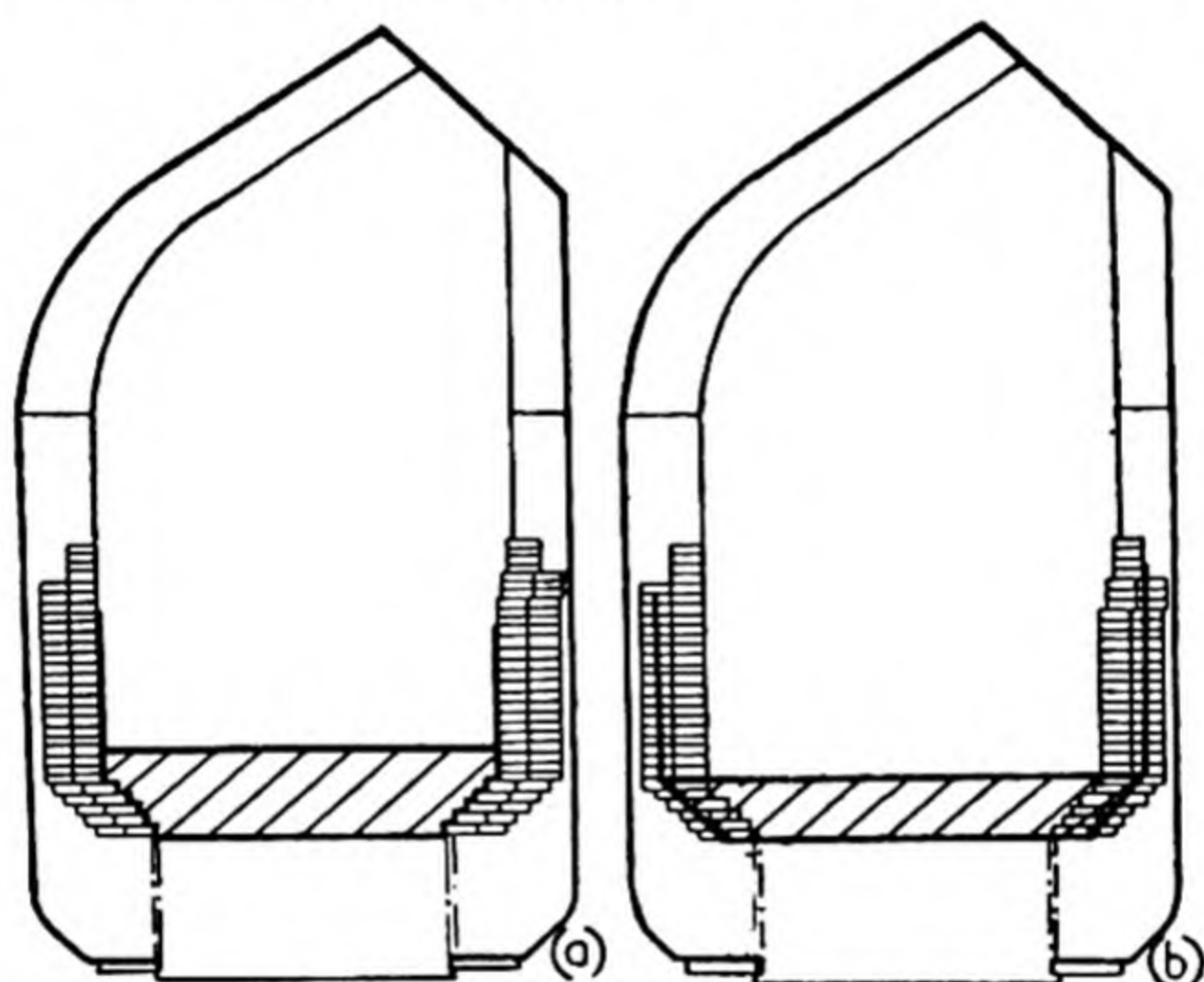


Fig. (1)7. Modern Bessemer Converter, with Shallow Bath, utilising new-type brick lining. (a) Start of First Bottom, Bath 24 in. deep, 9 ft. 6 in. wide max.: (b) Wear during Life, start of fourth bottom, Bath 16½ in. deep, 11ft. 6 in. wide max.

*(Courtesy of Iron and Steel Institute.)*

from a series of fine tuyeres in the base through molten pig iron so that carbon and silicon are removed by direct oxidation. At the same time the oxidation of these elements and of some iron liberates enough heat to raise the temperature from the melting-point of the original iron ( $1350^{\circ}\text{C.}$ ) to a temperature of the order of  $1600^{\circ}\text{C.}$ , well above the melting-point of the steel. Like Siemens, Bessemer developed this process to a commercial stage by setting up his own steelworks. The original Bessemer converter had an acid lining and so could only make good steel from a pig iron low in phosphorus and sulphur; one British steelworks has a supply of such pig and is operating the acid Bessemer. Sydney Gilchrist Thomas, with a remarkable insight into the physical chemistry of slag metal



reactions, developed the basic Bessemer or Thomas converter and greatly extended the range of ores the pig iron from which can be used for steelmaking in the converter. As in the basic open-hearth, the lining is of dolomite or magnesite and so it is possible to charge lime and operate with a slag which will absorb phosphorus and sulphur.

The present position of the Bessemer process *vis-à-vis* the open-hearth may be summarised as follows :

(1) The Bessemer process is less flexible than the open-hearth in the range of products obtainable. On the other hand, although the speed of operation of the converter is too great to allow the bath to be blown down to an exact carbon, by blowing down until the carbon flame disappears and then recarburising to the desired figure, the proportion of misfit casts obtained is at least as low as in the open-hearth. In the latter the carbon is adjusted by taking samples and tapping the furnace at the right moment.

(2) The Bessemer is unable to melt more than a very small percentage of scrap and in fact cannot melt the scrap produced in cropping and rolling its own ingots, so that it exports scrap. Since scrap is, in general, cheaper than pig, this puts up the cost of Bessemer steel. Looked at from the fuel point of view, the Bessemer uses no fuel itself but requires more coke to be used in blast furnaces.

(3) Bessemer steel has in the past had a high nitrogen content of the order of 0.015%, while the open-hearth furnace makes steel with a nitrogen content of 0.005%. High-nitrogen steels are advantageous for some purposes, such as wire and springs and free cutting steels, but are bad for deep drawing steels. However, for purposes other than sheets, which have to be essentially of the softest quality, high nitrogen steels have been made which have given excellent results after cold working and with a rigid control of grain size.

(4) The cost of compressing the air for blowing used to be rather high, although the use of modern turbo compressors can cut this down very much.

On the other hand, the advantages of the Bessemer are the low capital cost of plant, the very high production rate from a given plant and the possibility of synchronising ingot production with rolling mill requirements. Moreover, various processes have been worked out which can overcome the above-mentioned



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disadvantages. Thus the use of up to 40% oxygen-enriched air in the Bessemer gives the possibility of increasing the scrap which can be consumed, although this scrap will probably have to be of a physical form convenient for handling. Experiments<sup>1.19</sup> in England, France and Germany have resulted in the production of Bessemer steel with a nitrogen content as low as that in the open-hearth, although this puts up the cost of the Bessemer slightly. Dickie<sup>1.20</sup> describes methods involving the use of a shallower bath, based on the principle that the main object is to get the partial pressure of the nitrogen low and that this can be done by keeping the ferrostatic pressure low. The German method is based on the principle that so long as oxygen remains in the gas there will be little nitrogen pickup, and therefore the use of a very short path for the gases through the liquid metal is essential. It seems likely therefore that the Bessemer with oxygen-enriched blast or worked in conjunction with a cupola for melting scrap will probably play an increasing part in the steelmaking of the future.

The sideblown Bessemer or Tropenas converter, Fig. (1)8, also gives a low nitrogen content in the steel and may be developed in the future for large-scale basic ingot production, although in the past it has mainly been used for the manufacture of steel castings with acid-lined converters melting 1–5 tons and taking 30–40 minutes to do so. In this equipment a much lower blast pressure suffices to blow air down on to the surface of the molten pig, where the reaction takes place at the gas–metal interface. Experiments on the basic side-blown converter have shown that phosphorus can be blown out simultaneously with carbon. The use of an oxygen-enriched blast<sup>1.21</sup> in an acid side-blown converter has recently been shown to reduce the blowing time, give a higher steel temperature and reduce the silicon and other fuel consumption, while the fluid flow in such machines in relation to refractory erosion has been studied<sup>1.22</sup> by means of models, leading to suggestions for improved shapes.

Some of the higher-grade steels are still made in crucibles buried

<sup>1.19</sup> "Oxygen Enrichment in the Basic Converter," *I.C.T.R.*, p. 947 (21 Oct. 1949).

<sup>1.20</sup> DICKIE, H. A.: "Development of an Improved Basic Bessemer Steel," *J.I.S.I.*, **159**, Pt. 4, p. 360 (1948).

<sup>1.21</sup> HARRISON, J. L., NEWELL, W. C., and HARTLEY, A.: "Oxygen Enrichment in Side-blown Converter Practice," *J.I.S.I.*, **159**, Pt. 3, p. 282 (July 1948).

<sup>1.22</sup> NEWBY, M. P.: "Gas and Fluid Flow in a Sideblown Converter Model," *J.I.S.I.*, **162**, Pt. 4, p. 452 (1949).

in coke beds, Fig. (1)4, although Siemens showed that even for crucible melting the regenerative furnace results in a considerable fuel saving over the earlier methods.

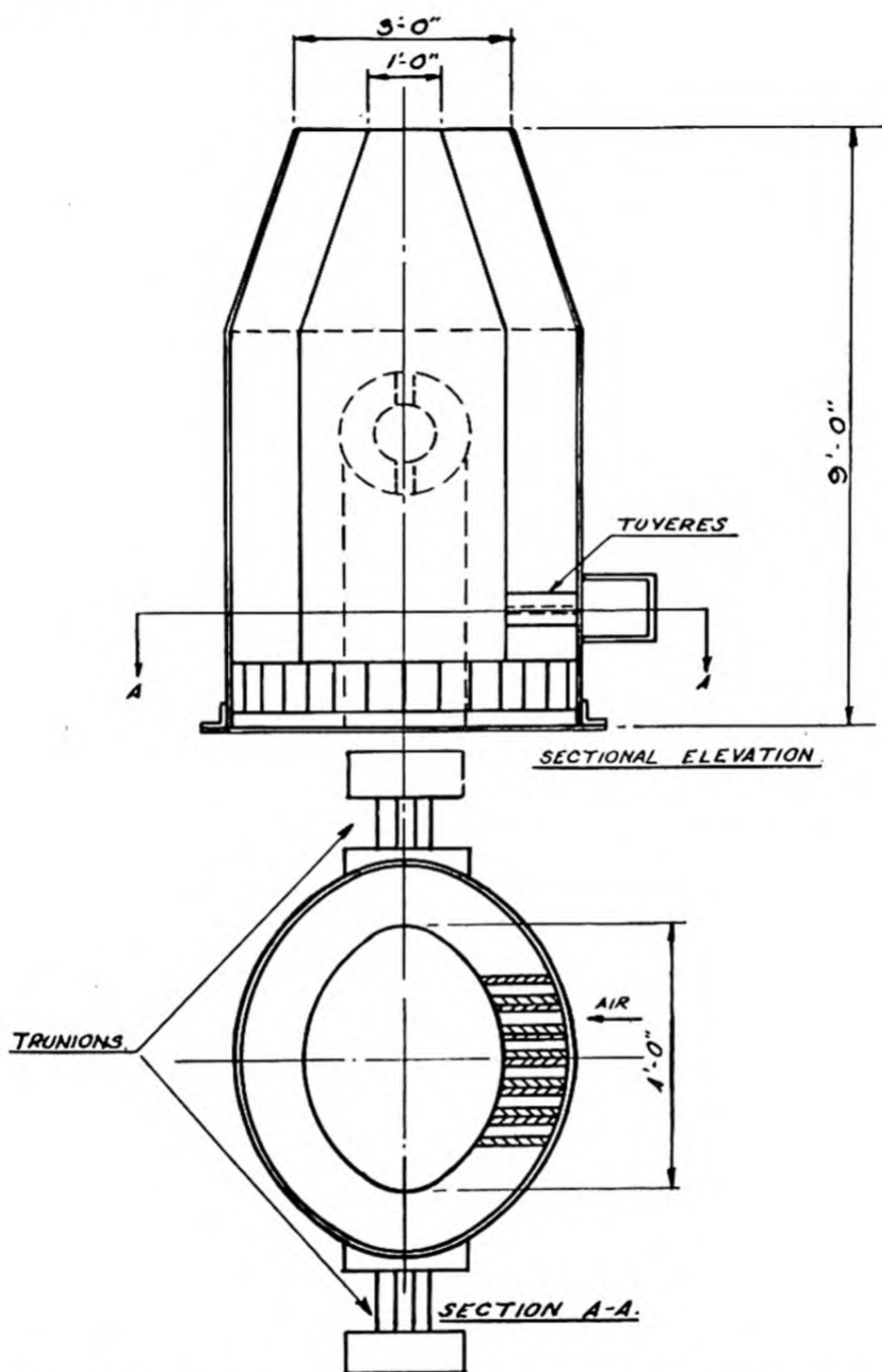


Fig. (1)8. Two-ton Side-blown Bessemer Converter for Steel Casting Manufacture.

The ingots cast from an open-hearth steel melt are withdrawn from the moulds as soon as they are sufficiently cooled to have a strong frozen crust and placed in a *soaking pit*, Fig. (1)9. The purpose of these furnaces is to achieve uniformity of temperature



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throughout the ingot at a figure ( $1250^{\circ}\text{C.}$ ) suitable for hot rolling. They are usually fired with hot producer gas, and air preheated either in recuperators or regenerators.

After the first rolling the bars are frequently cut up into shorter ingots, which are then reheated in smaller furnaces for further rolling to the final product (e.g. rod, sheet or angle iron). These furnaces may be of the continuous type, Fig. (1)10, in which material moves in counterflow to the heating gases, or, where the shape of material being heated is continually varying, they may be of the batch type.

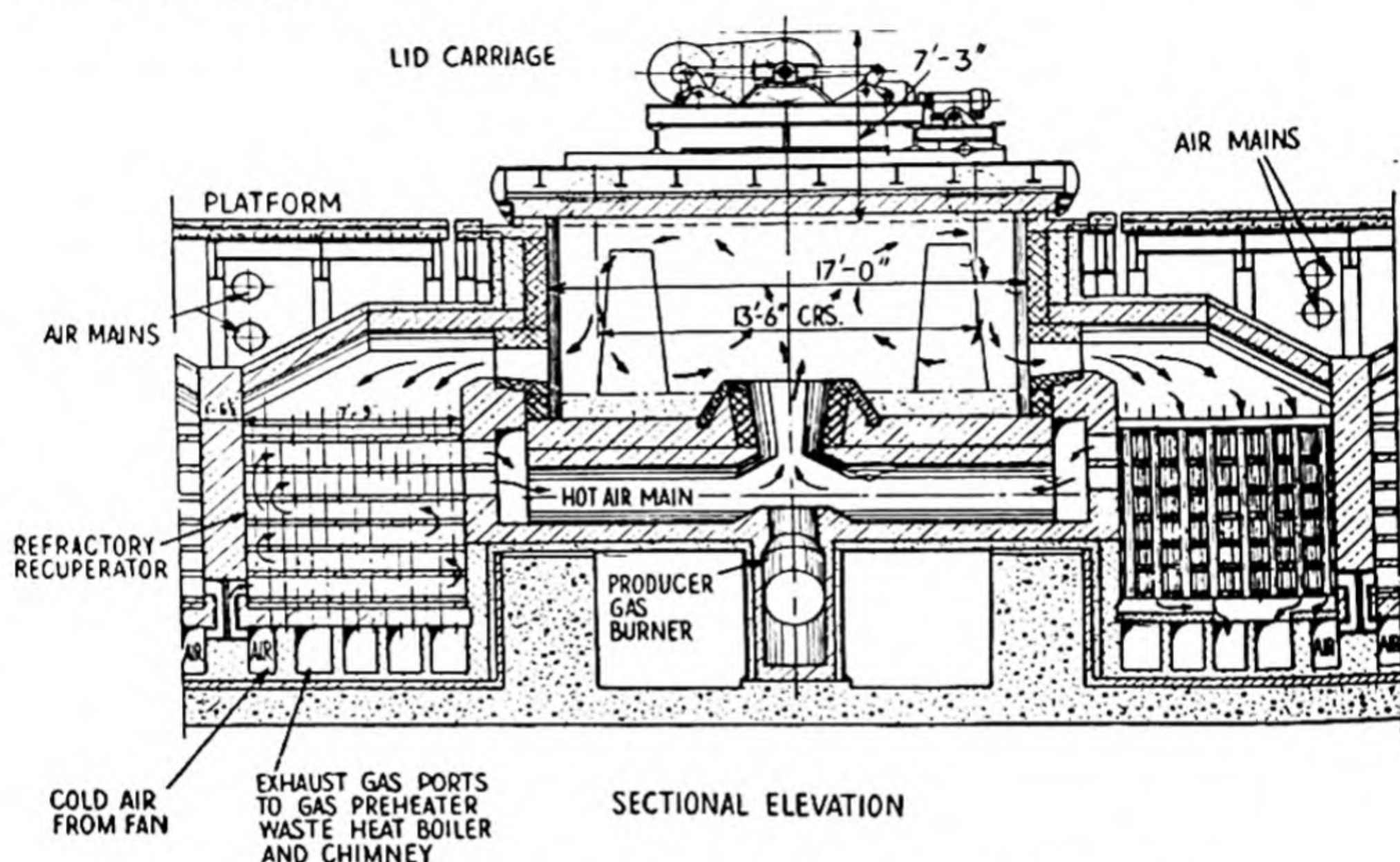


Fig. (1)9. Recuperative Producer Gas-fired Soaking Pit.

(Courtesy of the Wellman Smith Owen Engineering Corporation, Limited.)

The continuous furnace shown is of the pusher type, in which the blooms are pushed in at the right-hand end and this forces each successive one to move along the hearth to the far end, where they slide from the furnace. *The continuous pusher furnace* has been very extensively studied in Germany<sup>1.23</sup> and can give very good fuel economy (e.g.  $10^6$  to  $1.6 \times 10^6$  C.H.U./ton for  $1300^{\circ}\text{C.}$ )\* and a high output per unit of hearth area (50–100 lb./ft.<sup>2</sup>-hr.) because the heating rate at the colder end is very good and because the counter-flow principle can be fully applied. The main operating conditions

<sup>1.23</sup> HEILIGENSTADT, W.: "The Calculation of Push Furnaces," *Archiv. f. d. Eisenhüttenw.*, 7, pp. 25, 103 (1933, 1934).

\* Theoretical requirements 450,000 C.H.U./ton.

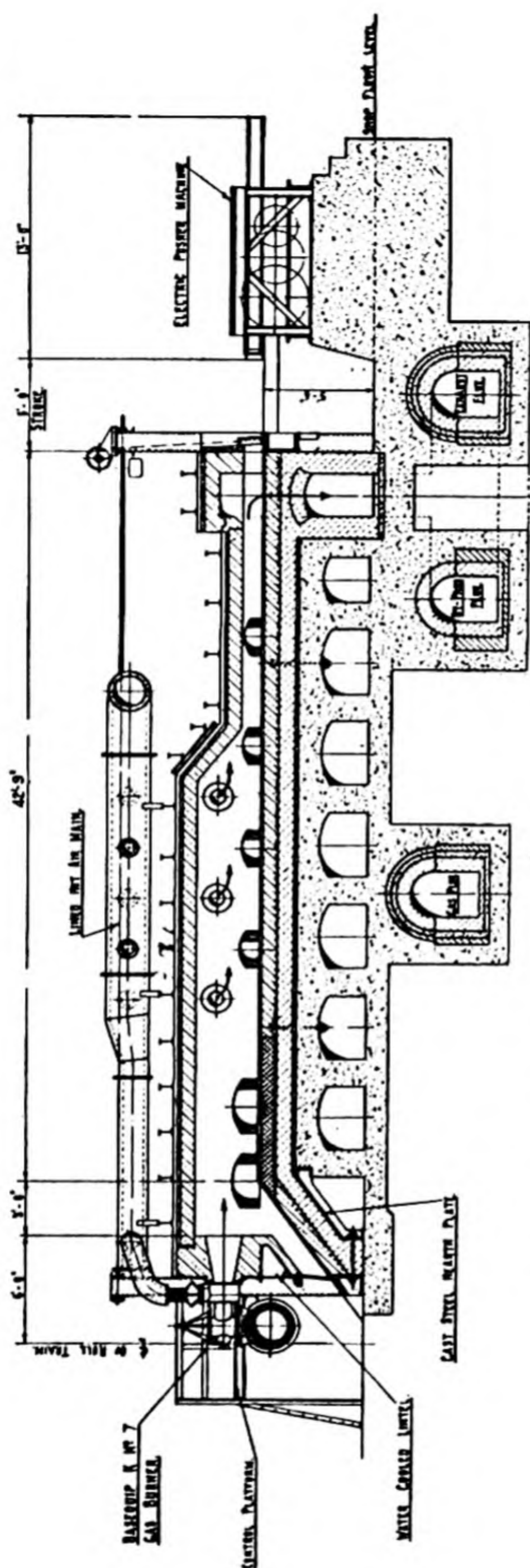


Fig. (1)10. Section of Continuous Pusher Furnace for Reheating Steel Slabs.  
(Courtesy of the Wellman Smith Owen Engineering Corporation, Limited.)



### 1.2.2 THE SCIENCE OF FLAMES AND FURNACES

which must be satisfied if good fuel economy is to be obtained are that the heat transfer area should be sufficient for the gases to be brought down to  $700^{\circ}$ – $800^{\circ}$  C. in the furnace itself before passing to a recuperator and that the pressure in the furnace should be so adjusted that very little air is drawn in through the ports and the charging end. The chief disadvantage is the necessity to use water-cooled skids to protect the hearth from abrasion. Pusher furnaces are fired with pulverised coal, producer gas, mixed gas or coke oven gas.

A recent design of small gas-fired continuous billet-heating furnace is shown in Fig. (1)11.<sup>1.24</sup> Continuous furnaces of the bogey type, in which the billets are fed along on refractory-lined

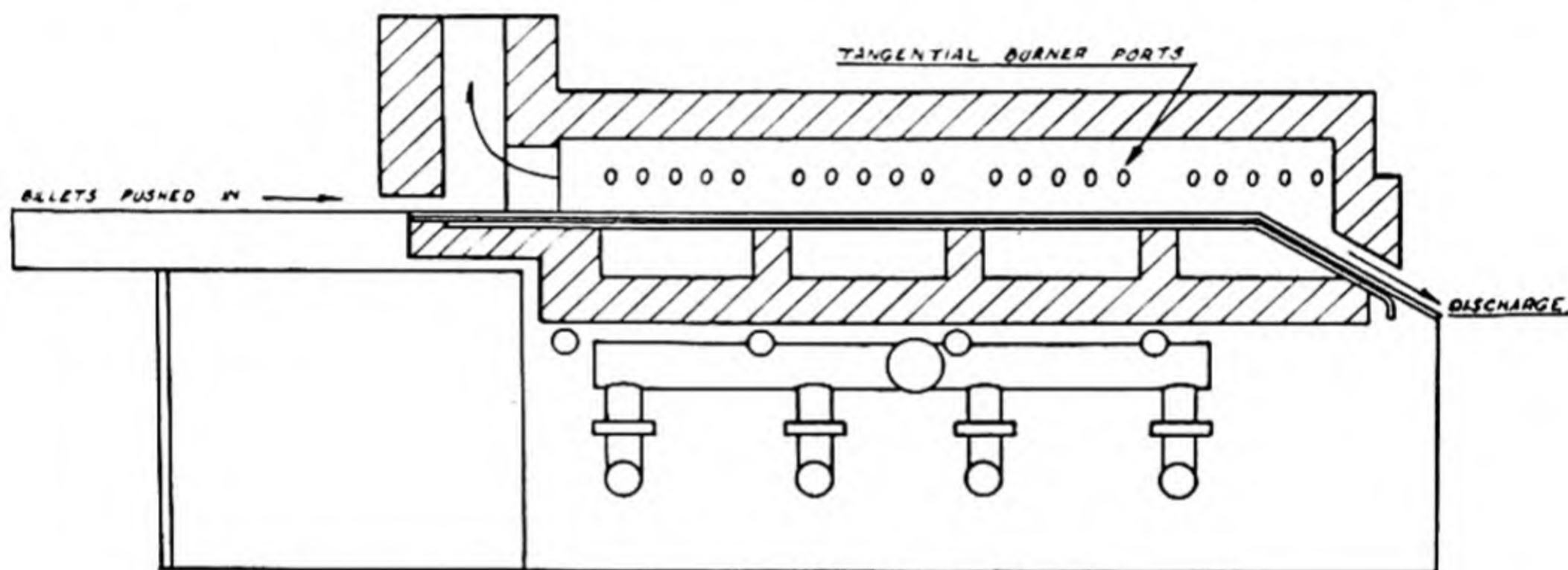


Fig. (1)11. Gas-fired Continuous Billet-heating Furnace. (Rapid Billet-heating with Gas.)

bogies forming a moving floor to the furnace, are also used. Batch furnaces may be fired by producer gas, pulverised coal, or, more usually, by built-in fire-boxes with deep fuel beds so that they act as semi-producers giving long flames with secondary air. The direct-fired batch-heating furnace is subject to the objection that it rarely makes much use of the heat in the gases leaving the furnace because the air to the fire cannot be preheated.

**1.2.3. Non-ferrous Metals.** Many of the furnaces used in the non-ferrous metals industries are similar to those used in the iron and steel industries ; for example, vertical shaft kilns similar to the blast furnace are used for the reduction of copper ore. In view of the lower temperature required for melting, however, the bath-type furnaces ("reverberatory furnace") used for melting and preparing copper and its alloys such as brass, are fired with pulverised coal or

<sup>1.24</sup> STANBERY, L. J., and BRENNAN, J. M.: "Rapid Billet Heating with Gas," *Iron Age*. (26 Aug. 1948).



direct-fired, and regenerative preheating is rarely used. Open-hearth melting furnaces used in the copper industry may contain very large baths (e.g. 300 tons) of metal, Fig. (1)12, although brass is often made in 1-ton baths.

In the manufacture of copper,\* roasted sulphide concentrates (containing around 30% Cu) are charged to a large reverberatory furnace 35–40 ft. across, 120–130 ft. long, 12 ft. high at the arch and with 200–300 tons capacity. Here the concentrates are melted and partially refined by a slag made with silica lime and iron oxide. The resulting material, called a matt, which is largely iron and copper sulphides, is then blown in large horizontal converters which

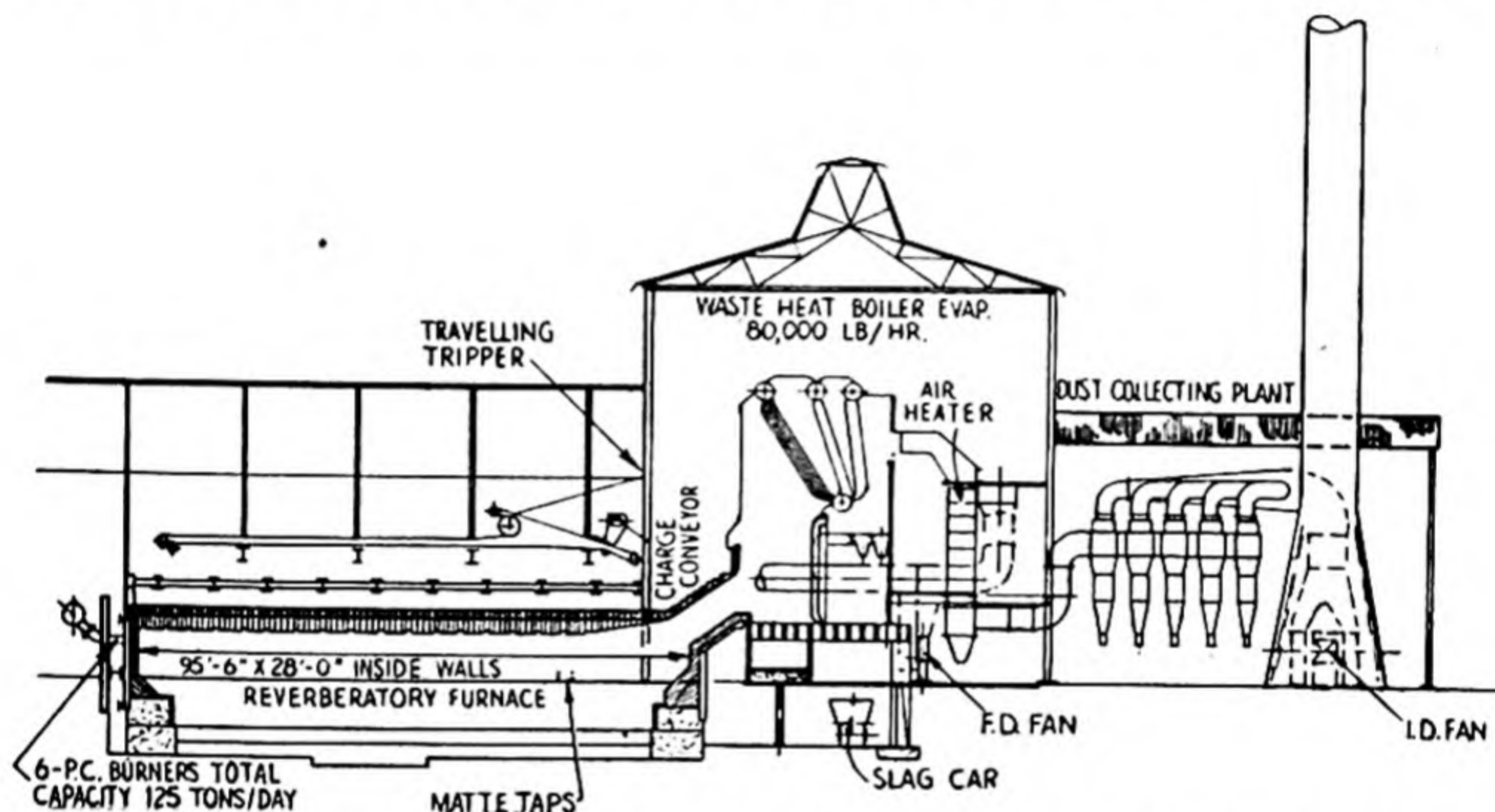


Fig. (1)12. Longitudinal Cross-section through Reverberatory Furnace, Waste-heat Boiler and Dust-collecting plant as employed in the Copper Industry.

(Courtesy of The Roan Antelope Copper Mines, Limited.)

oxidise sulphur and iron to  $\text{SO}_2$  and iron silicate respectively, the Cu-rich slag resulting from this blowing going back to the first reverberatory furnace. The blow is in two stages; the first removes the iron, the second the sulphur, and the resulting material, which is around 99% Cu, is called blister copper. The blister copper then goes to a second reverberatory furnace, where it is remelted and further oxidised by the furnace atmosphere and in some cases by blowing through iron pipes with the inner ends submerged. This results in a further purification, particularly elimination of iron, but also oxidises some copper, putting up the oxygen content of the metal to 0.5–1.0%. The next stage depends on the

\* For a thorough discussion of copper furnace practice see *Trans. Amer. Inst. Mining and Met. Eng.*, 106, "Copper Metallurgy" (1933).



### 1.2.3 THE SCIENCE OF FLAMES AND FURNACES

character of the original ore : if it contained no Ni, Bi or Ag then the metal from the second reverberatory furnace is sufficiently pure for many purposes and is called "fire refined" and is cast as wire bar. If these elements are present the metal is cast as anodes and then electrolysed in a sulphate bath ; now it contains only hydrogen and sulphate as impurities. It is remelted a third time in a reverberatory furnace under oxidising conditions, the slag is skimmed off and it is reduced back by stirring it with green telegraph poles. It is then ready for the final casting and rolling.

Rotary furnaces, Fig. (1)13, are frequently used for melting

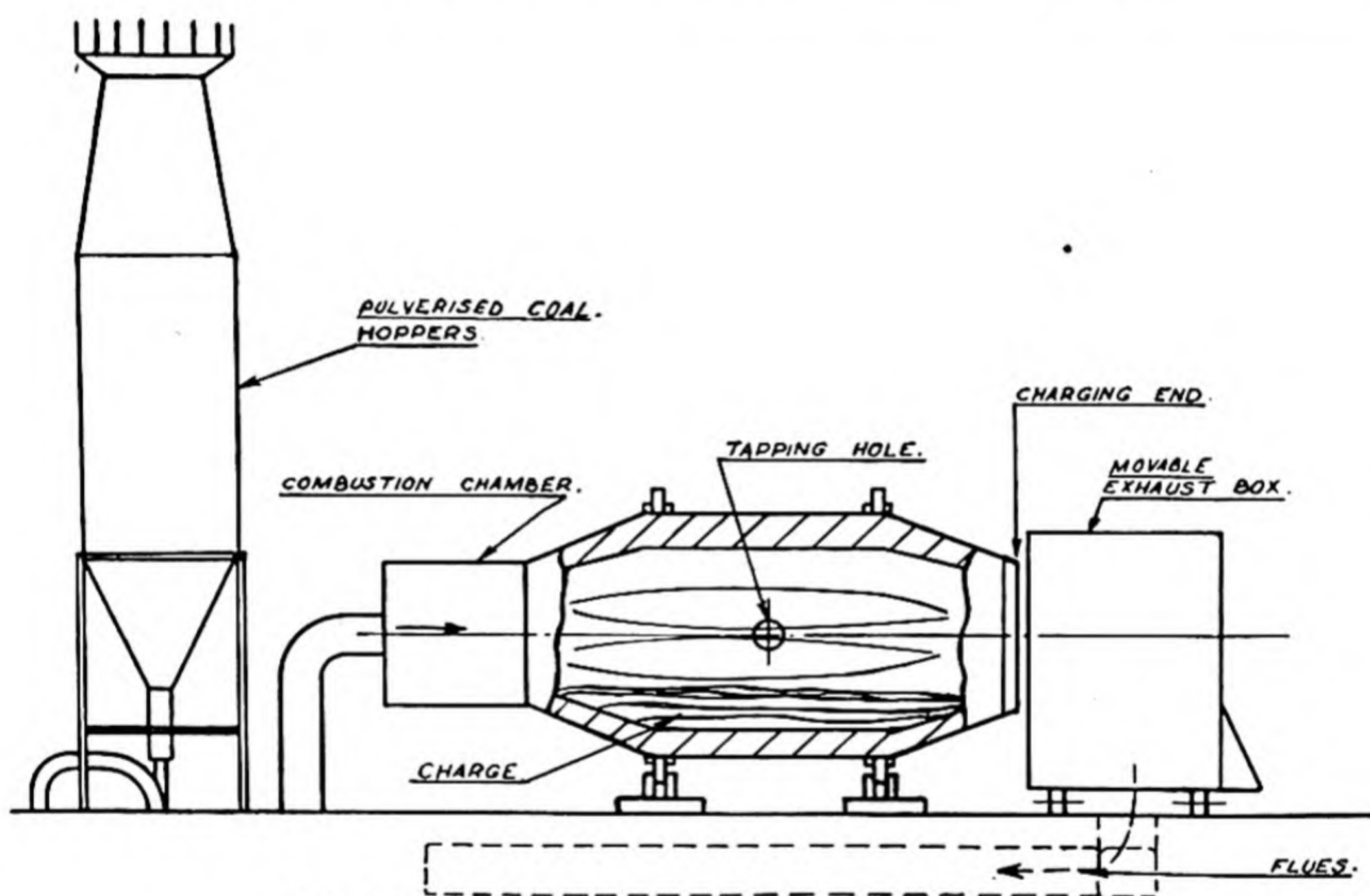


Fig. (1)13. Small Rotary Brass Melting Furnace.

non-ferrous metals, having the advantage that heat given to the crown is continually carried under the molten metal so that the latter is really heated from below as well as from above, while the crown is continually cooled.

For smaller-scale melts the crucible furnace, Fig. (1)14, is very popular in the non-ferrous industry, the crucibles in this case being fired with gas or oil more frequently than by direct contact with a bed of coke. Such furnaces rarely show a thermal efficiency above 10% even when arrangements are made for preheat, as very little of the heat of the gases below the crucible temperature is used. Designs are usually prepared with the aim of obtaining a high

convective heat transfer by using tangential inlets to establish a rotary gas flow around the crucible.

**1.2.4. Refractories, Pottery and Brick.** For many of the materials fired in these industries the net heat requirements are very small, since the material gives up nearly as much useful heat in cooling down as is required to heat it up. Consequently, the older furnaces of the batch type, where no use whatever is made of the heat given up on cooling, have a fuel requirement very much greater than the more modern types. One form of the older batch-type furnace

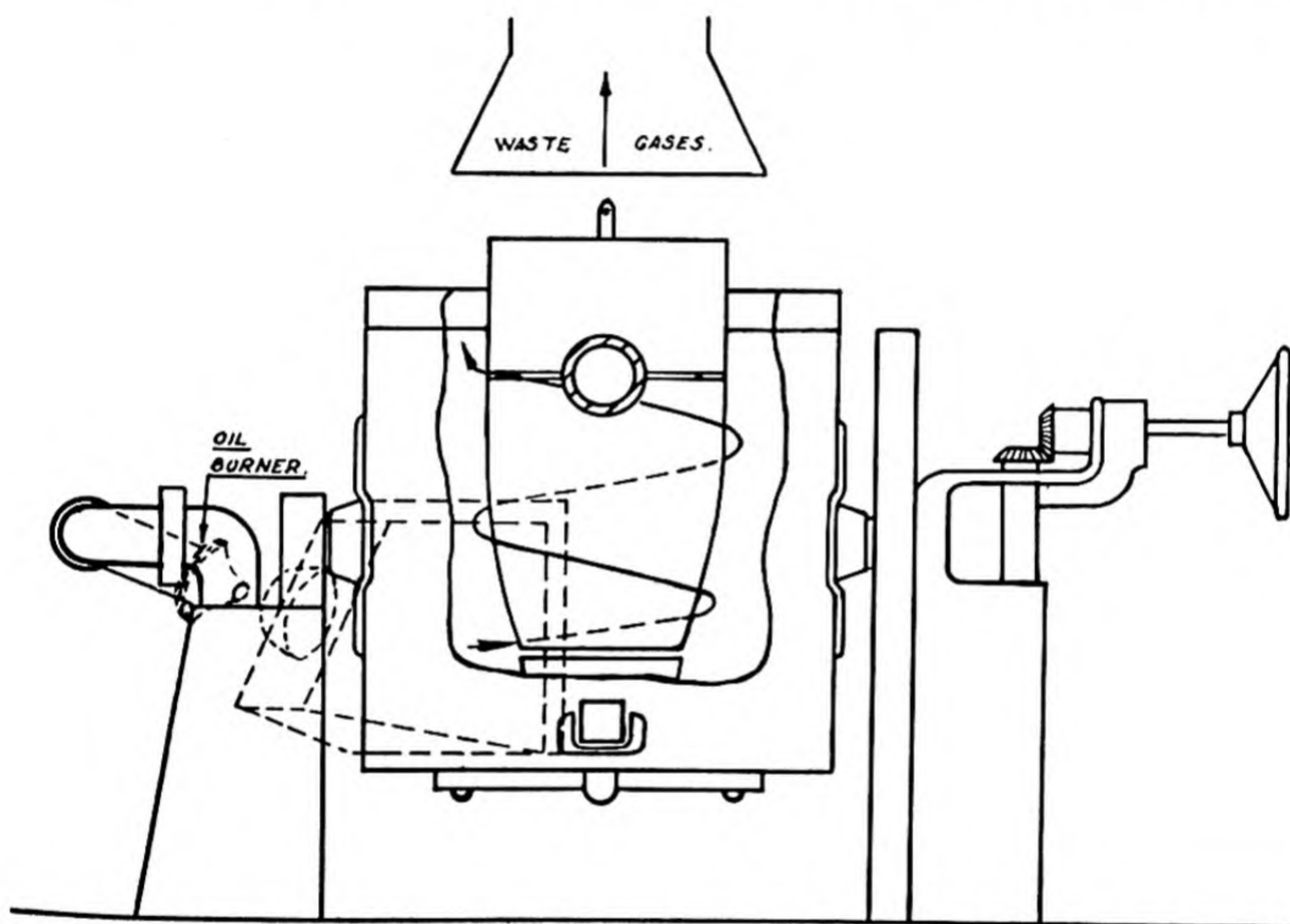


Fig. (1)14. Oil-fired Tilting Crucible Furnace for Non-ferrous Melts.

(Courtesy of The Morgan Crucible Company, Limited.)

(the "beehive kiln") consists of a roughly hemispherical chamber fired by means of a number of fuel beds around the circumference, the gases passing up the inside of the wall, see Fig. (1)15, down through the ware, which is piled with spaces between, and out through holes in the floor which are connected to the flue.\* One more modern design of this type of furnace is half-cylindrical instead of hemispherical, and fired with producer gas from a row of ports

\* It was shown by Groume-Grjmailo (see 5.6.3) that *downward* flow of gases hotter than the surroundings gives an arrangement in which convection assists the uniformity of gas distribution across the wares.



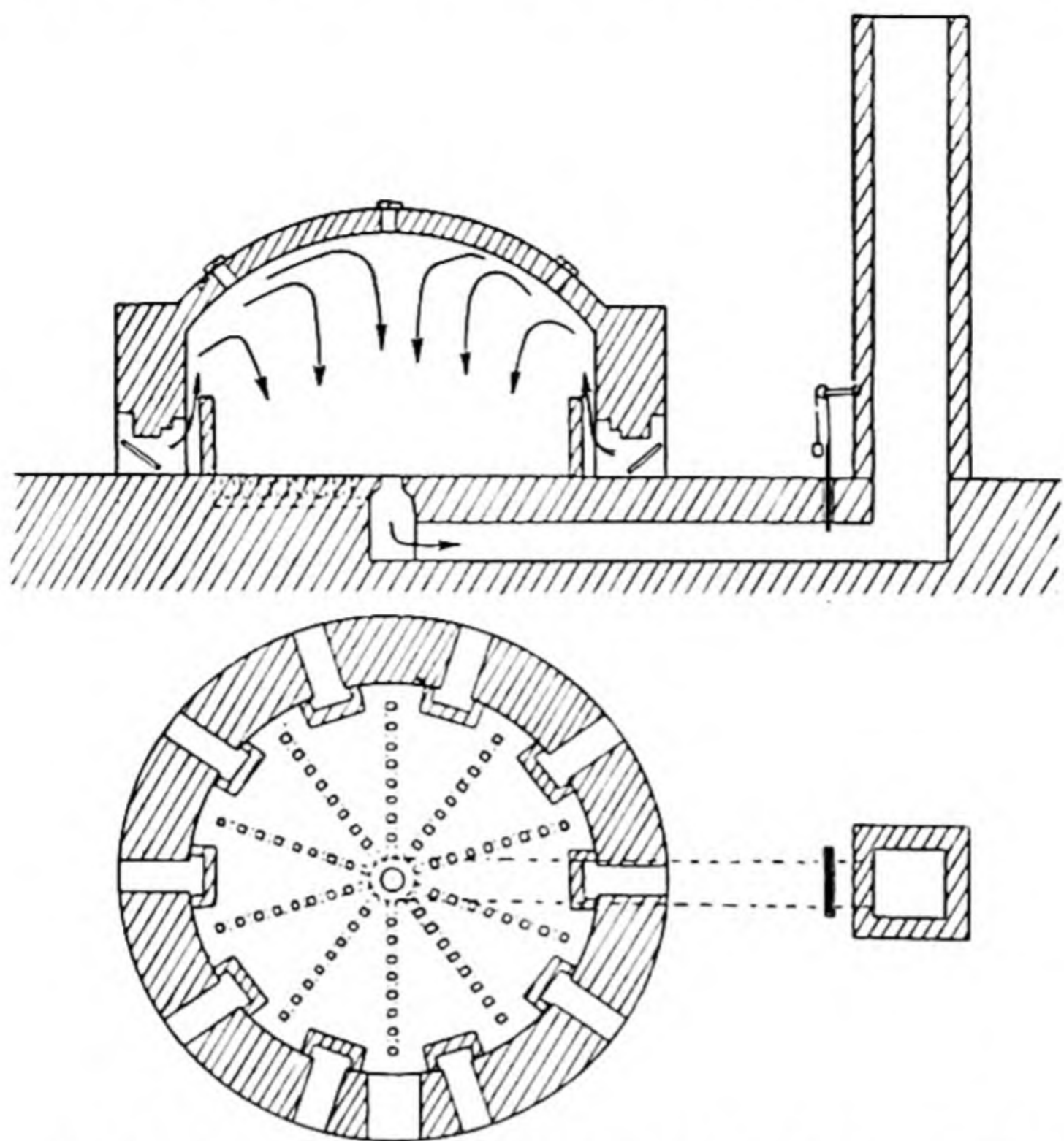


Fig. (1)15. Coal-fired Beehive Kiln for Refractory Firing.  
(Courtesy of British Ceramic Research Association.)

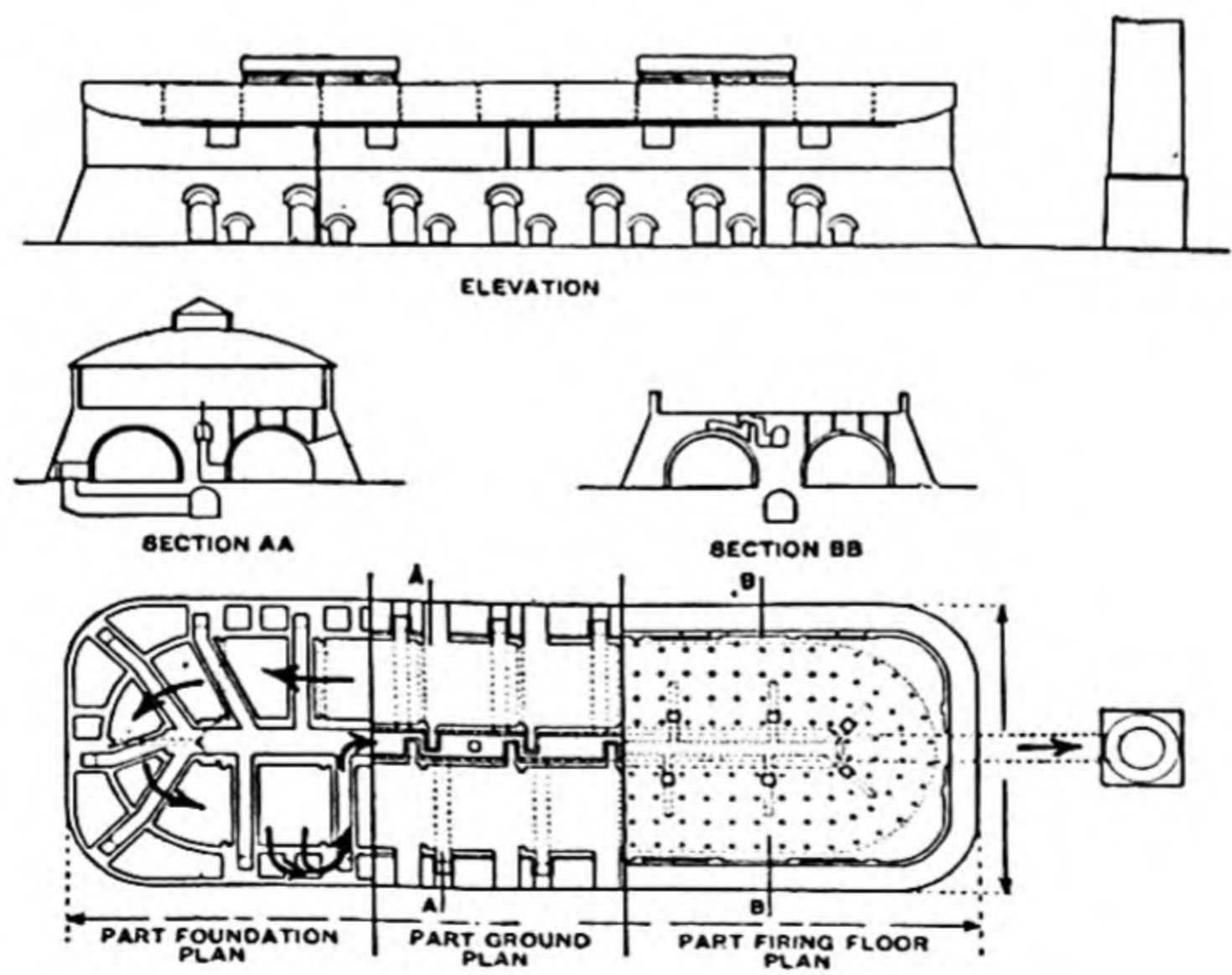


Fig. (1)16. Semi-continuous Brick Kiln.  
(Courtesy of The British Ceramic Research Association.)

along each side. The waste heat of the gases leaving the firing chamber can then be used for preheat of the secondary air or for waste heat boilers, but it is still difficult to make use of the heat given out by the ware on cooling down. Beehive kilns are still used for firing silica bricks at temperatures up to  $1500^{\circ}\text{C}$ . as well as for pottery firing. Similar furnaces were formerly in use for coke making and for glass melting in pots.

Improved kilns make more use of the heat given up by the ware during cooling in two ways. The semi-continuous kiln, Fig. (1)16, has a number of chambers, the actual firing being applied to each one in turn. The combustion air is preheated by passing it through the preceding chambers in which the ware is cooling ; combustion

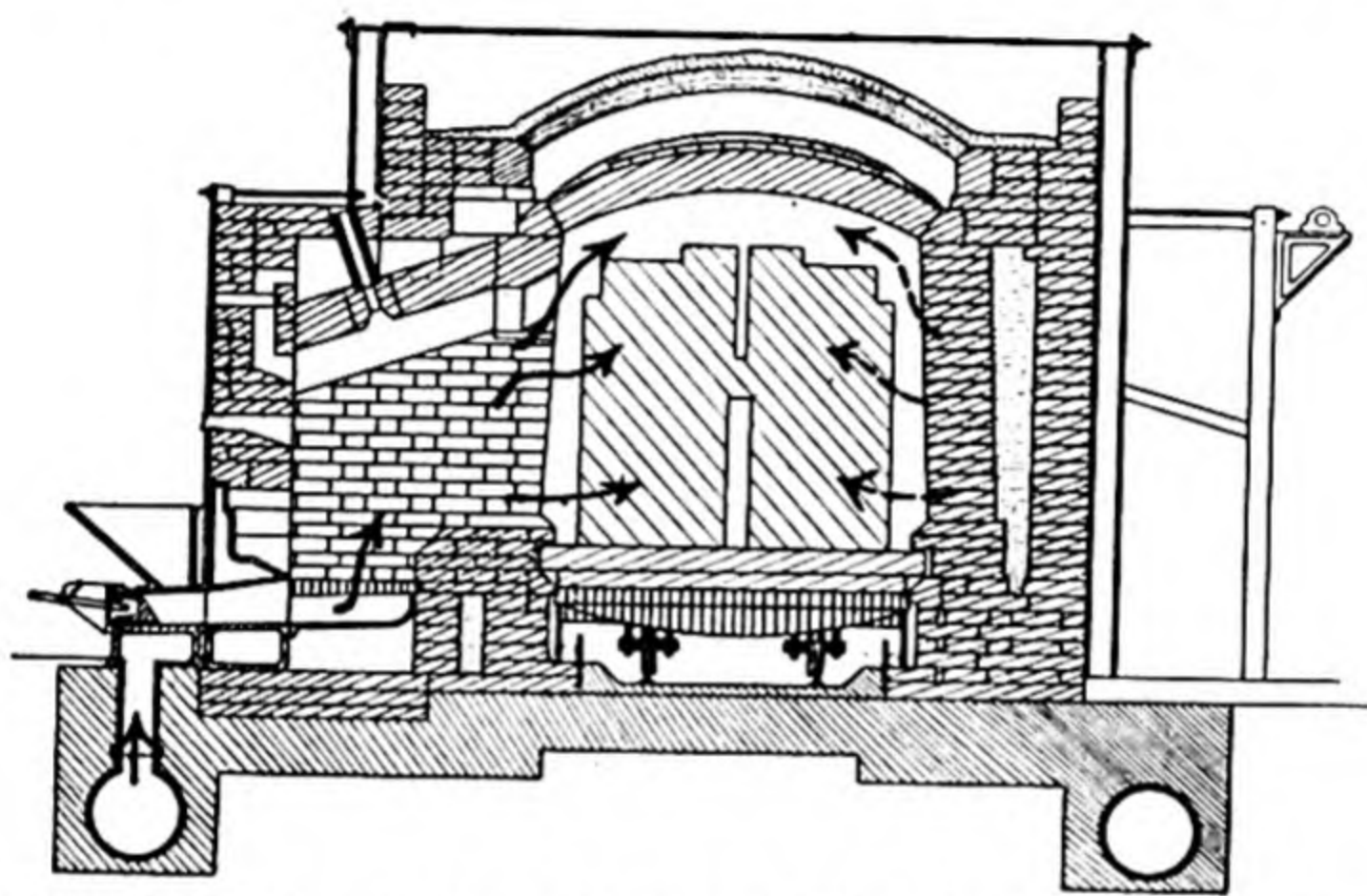


Fig. (1)17. Sectional view of Tunnel Kiln for Refractory Firing.  
(Courtesy of The British Ceramic Research Association.)

takes place in the hottest chamber and the gases leaving this one heat the ware in successive chambers to successively lower temperatures, the last chamber being connected to the stack. Finally, there is no flow at all in one or more chambers opposite to the hottest one where the ware is being removed and recharged. When the firing of the hottest kiln is completed the fire is moved to the next one, and the dampers are altered so that each chamber is now in a position corresponding to that previously occupied by the one after it on the gas path.

Tunnel kilns, Fig. (1)17, and continuous circular kilns, use the same method of preheating the combustion air by the cooling ware and counterflow heating as the semi-continuous kiln, but the ware is now moved on trolleys or a platform through successive zones instead of the combustion zone being moved round a static kiln



#### 1.2.4 THE SCIENCE OF FLAMES AND FURNACES

system. Silica bricks are sometimes fired at  $1500^{\circ}\text{C}$ . in tunnel kilns, and such kilns may also be used for calcining or sintering raw materials such as magnesite or sillimanite.

Both the semi-continuous and the tunnel kiln make the exact control of the firing stages of each individual batch more difficult than it is with the batch kiln, since the same gases are being used to heat different batches which are at different stages of the cycle. There is therefore to some extent a choice between obtaining the minimum fuel consumption in the furnace and close control of the firing conditions. The use of the more efficient high-throughput continuous furnace is therefore closely linked with the uniformity of the batch materials and of the desired product, and fuel economy can be obtained only by obtaining uniformity by means not requiring high-grade heat.

In firing pottery the ware is frequently protected from the combustion gases by enclosing it in muffles or in separate fireclay or silica containers, which are piled one on another like the baskets of a Covent Garden porter, known as "saggers." The effect of such muffles is greatly to increase the fuel consumption, for reasons which will be discussed in Chapter 2.

Building bricks,<sup>1.25</sup> Fig. (1)16, are fired mostly in the semi-continuous type of kiln, the chief difference from the kilns used in the refractory industry being that the fuel, in the form of  $\frac{1}{2}$ -inch slack, is actually fed down through "chimneys" left in the stacks of unfired bricks and burnt by means of air going up through a grate at the bottom. The firing temperature in this case is only about  $900^{\circ}\text{C}$ . Where the clay contains an appreciable amount of combustible (coal or oil) the fuel supplied separately can be greatly reduced. In Germany bricks are often made with a proportion of very fine brown coal dust mixed with the clay.

In the pottery industry the cost of fuel is frequently very small compared with the value of the ware fired, so that fuel economy must take second place to close control of firing conditions. In the brick industry on the other hand, fuel costs are a large item and careful attention is given to choosing firing methods which are most economical of fuel.

**1.2.5. The Chemical Industry.** The furnaces used in the manufacture of heavy chemicals are very varied, but one characteristic

<sup>1.25</sup> NOBLE, W., ROWDEN, E., and BOOTH, N.: "The Utilisation of Small Coals in the Firing of Common Building Bricks," B.C.U.R.A. Conf., 1943.



type is the modern development of the witches' cauldron, Fig. (1)18, consisting of a large cast-iron vessel heated from below by flames and frequently also by direct radiation from a fuel bed. Direct radiation is probably of great importance in such furnaces,\* as it is difficult to obtain a good scrubbing action of the gases on the outside of the vessel. Just as in boiler firing, the heat is required at a temperature much below the combustion temperature, and so the first law thermal efficiency is high although there is really a serious degradation of energy (see Chapter 2).

Tar stills provide another example of this type of furnace.

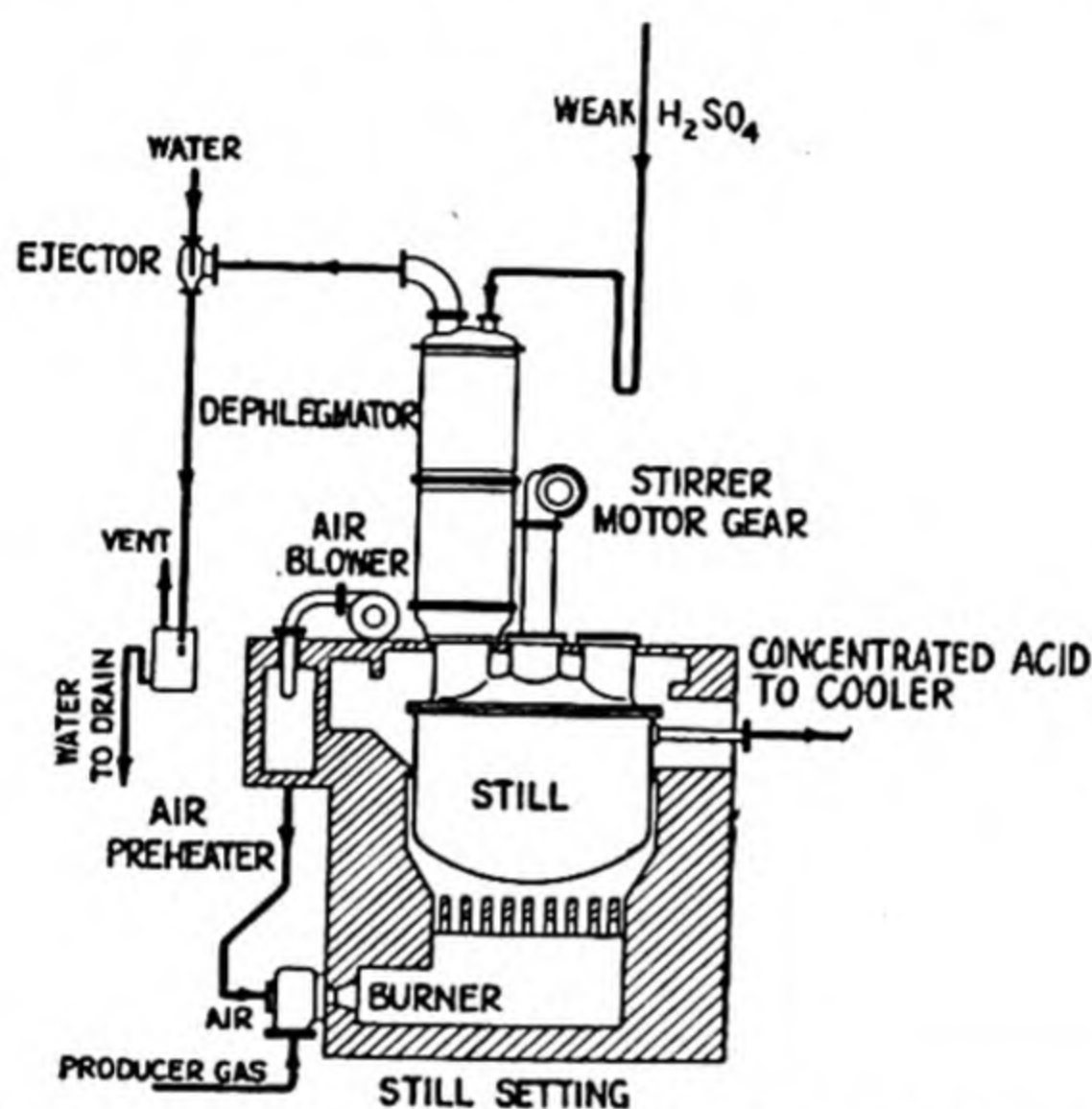


Fig. (1)18. Cross-section of Sulphuric Acid Concentration Still.

(Courtesy of Bamag, Limited.)

**1.2.6. Cement Lining.** Cement is made, almost without exception, by feeding a wet slurry of ground chalk and clay into the upper end of a gently sloping rotary kiln, Fig. (1)19, which may be anything up to 300 ft. long and 10 ft. in diameter. The kiln is invariably fired with pulverised coal at the lower end, this type of firing being suitable for two reasons. In the first place it gives a long flame and hence a large region of uniformly high temperature in which the final calcination can take place. Secondly the ash from the coal finds its way into the cement, where it does no harm as its composition is chemically similar.

There are two main advantages in a rotary kiln of this type as compared with a static bed of material over the top of which the

\* GRUMELL, E. S.: Private communication.



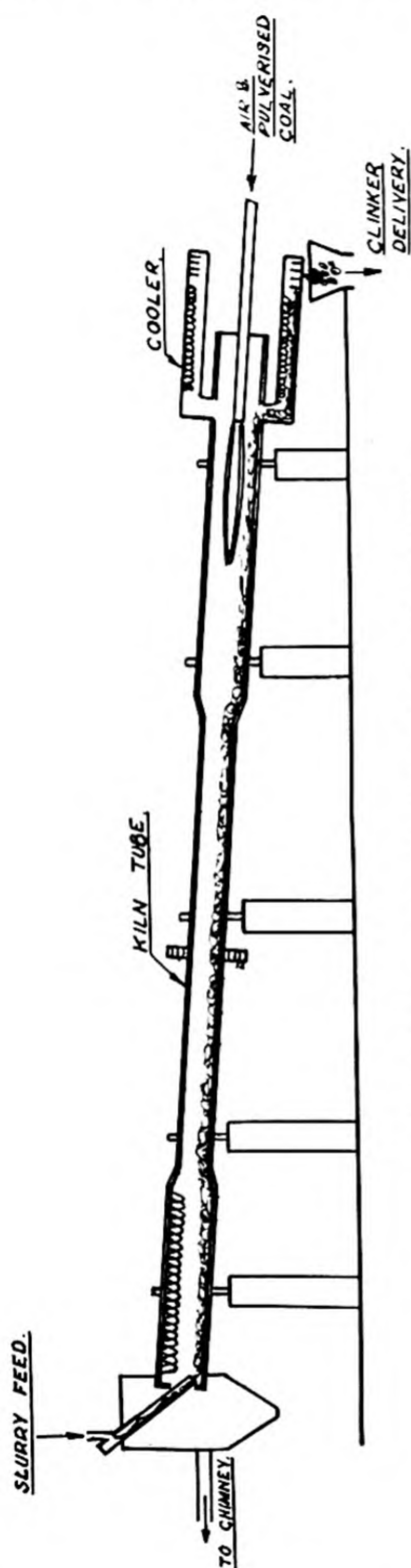


Fig. (1)19. Pulverised Coal-fired Rotary Cement Kiln.

heating gases pass. One is that the part of the brickwork out of contact with the material is continually receiving heat and carrying it underneath the material, as was discussed in connection with rotary melting furnaces. Still more important, however, is the way in which a rotary kiln causes the fine material to fall continually through the flame, so picking up heat by convection to a very large exposed surface area. The rotary kiln is thus ideally constituted for heating fine materials (e.g. less than  $\frac{1}{2}$ -inch diameter) which do not soften.

On the other hand, the rotary kiln has the disadvantage that it is very difficult to insulate the brickwork, since any soft material interposed between the hard firebrick and the outer shell is liable to be pulverised by the continuous movement. For this reason arrangements are sometimes made to recover some of the heat losses by placing air pipes carrying secondary air in a hairpin loop above the kiln. This method of heat recovery has, however, a theoretical disadvantage compared with the use of heat in the waste gases, as in Chapter 2. In some rotary kiln systems the hot products coming from the lower end of the kiln are allowed to drop into a further independent rotating section, where they are cooled by passage along in counter current to incoming

secondary air. This form of preheat is thermodynamically sounder. The rotary kiln is unsuited to deal with material which softens on heating, as this forms a ring which eventually closes the furnace right



up. Magnesite and other refractories required in granulated form are often burnt in rotary kilns.

The other form of kiln used in these industries is the vertical shaft kiln, Fig. (1)20. These kilns somewhat resemble a blast furnace or cupola. In some types, as for example those used for sintering the dolomite which is used for fettling open-hearth furnaces, the raw material is charged in layers alternating with layers of coke, the combustion of which by means of air blown in through tuyeres provides the heat. These types of kiln can be used to give temperatures of up to  $1500^{\circ}\text{C}$ ., but the material which comes out is

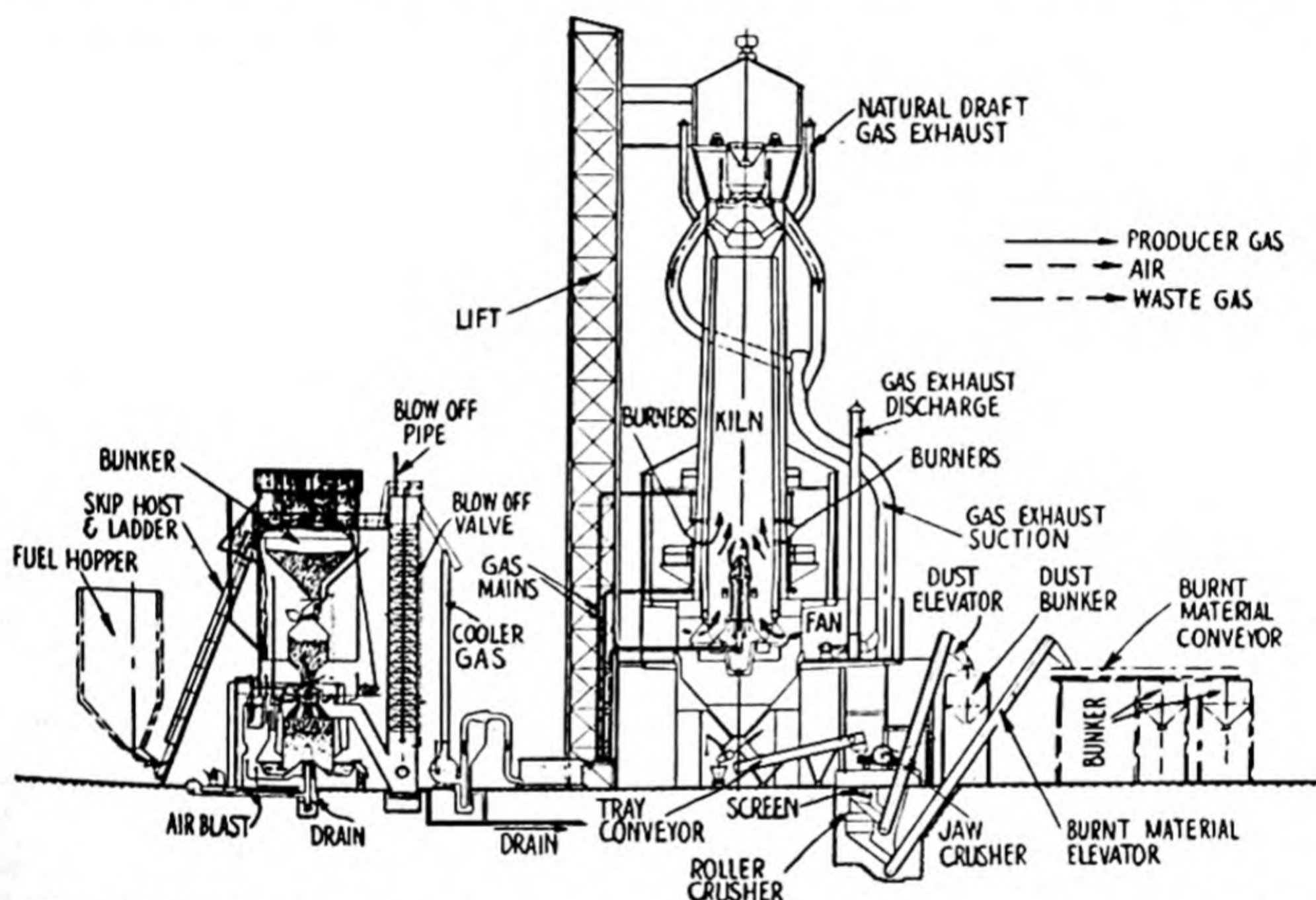


Fig. (1)20. Diagrammatic arrangement of Gas-fired Mechanically Charged and Discharged Dolomite Plant.

(Courtesy of the Power Gas Corporation, Limited.)

necessarily contaminated with the coke ash. Where purer products and lower temperatures are required, as in the manufacture of reactive burnt dolomite, this type of kiln is fired with producer gas made from coal or coke in separate producers. In this case it is necessary in the larger-diameter kilns to have a central burner to avoid under-firing the centre. Some of the air is passed in near the smaller burners on the outside wall and some is allowed to flow in through the discharge mechanism at the base, receiving heat from the cooling products before it arrives at the combustion zone. It is possible to operate these producer-gas-fired kilns so as to give no



## 1.2.6 THE SCIENCE OF FLAMES AND FURNACES

excess carbon monoxide, whereas the ones into which coke is fed in with the charge always make a kind of low-grade combustible gas similar to blast-furnace gas.

**1.2.7. The Glass Industry.** Glass is made by melting silica sand, soda, lime, broken glass and other constituents in various types of furnace, the greatest bulk producer being the open tank, while more expensive glasses are mostly made in pot furnaces. The open tank, Fig. (1)21, is somewhat similar to the open-hearth furnace, having been designed, like it, by the Siemens family in the period 1850–1870 ; it works, however, at an appreciably lower temperature. All the larger glass tank furnaces are regenerative. With producer gas firing, a pair of regenerators is situated along

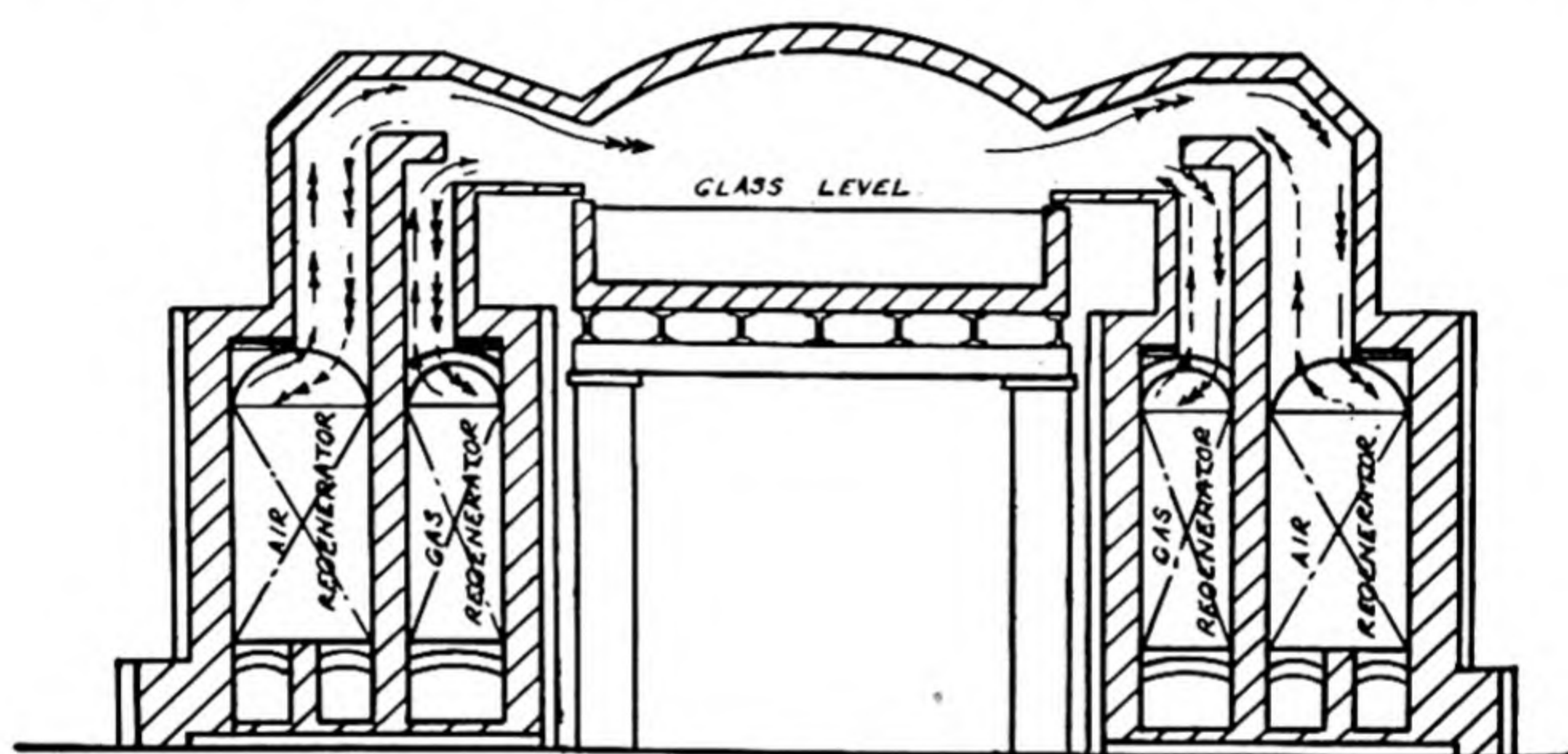


Fig. (1)21. Regenerative Glass Tank Furnace.

each side of a rectangular bath, each pair being connected in parallel to three, four or five ports on that side. Oil firing or rich gas as in the open-hearth requires only one regenerator on each side. The gases thus flow alternately in either direction across the shorter side of the rectangle. In the glass industry the reversals usually take place at regular half-hour intervals and the regenerators are much larger per unit of fuel burnt than those of the steel industry. If it were not for the difficulty of ensuring uniform distribution of the gases at such low velocities the regenerators would therefore give a very high measure of heat return. In some cases regenerators with horizontal flow in two or three passes are used to ensure utilisation of all the brick volume.

Producer gas is the most usual fuel, and as the surface of the glass is exposed to the flames the appearance in the flame of particles of



ash from the producer is very undesirable, since a minute percentage of iron will colour the glass. The gas checkers appear to be effective dust catchers in many cases. Oil and creosote pitch with pressure or steam atomisation are also used in some furnaces. This type of furnace works at temperatures between  $1300^{\circ}$  and

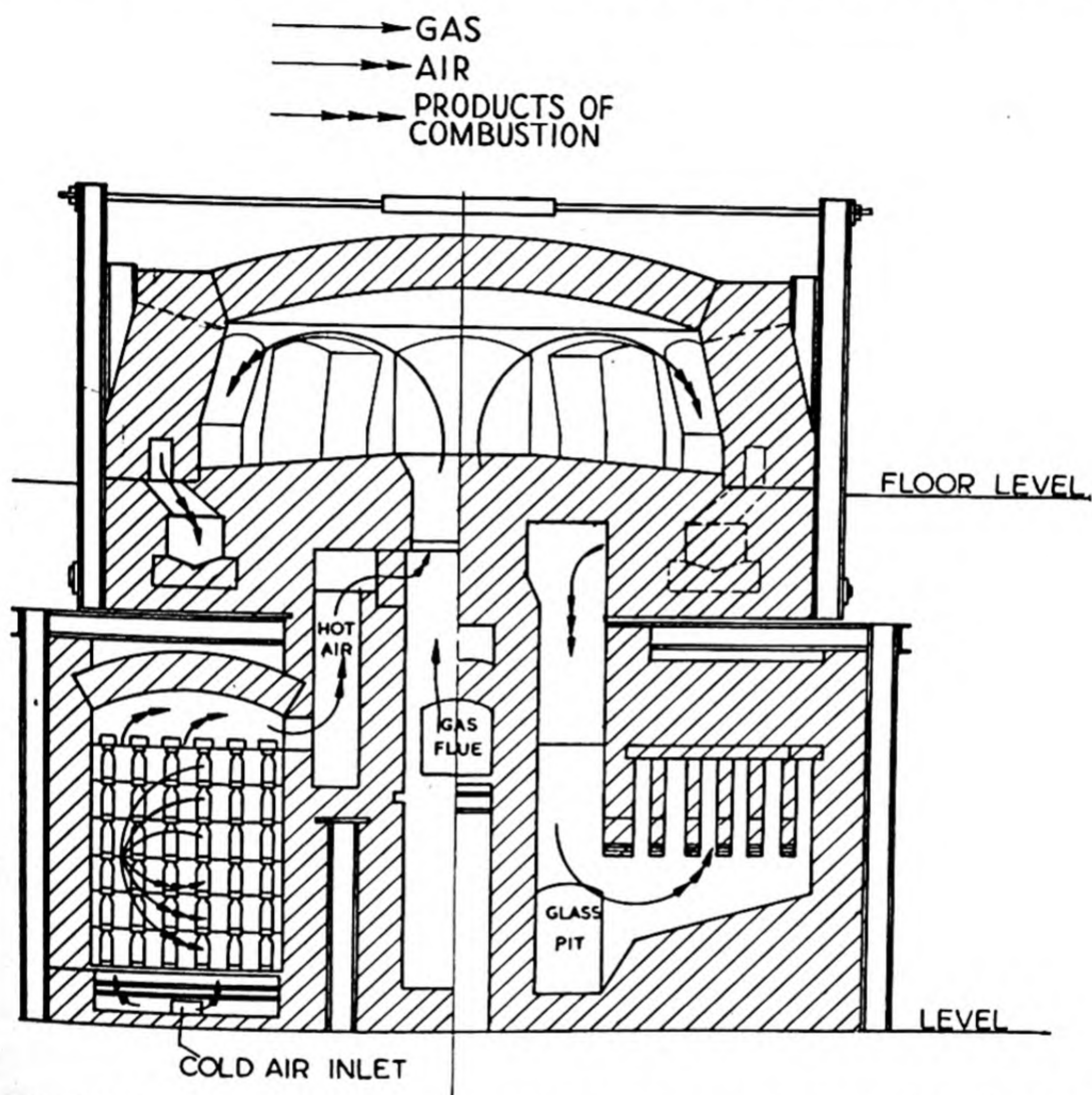


Fig. (1)22. Cross-section of Producer Gas-fired Recuperative Pot Glass Furnace  
(Courtesy of Stein & Atkinson, Limited.)

$1500^{\circ}$  C., the higher one applying to the glasses of high silica content (e.g. Pyrex), and it carries out a triple process of melting, mixing and allowing bubbles to escape from the batch materials. Such furnaces frequently run for 1–2 years continuously without shutting down, batch being fed in at one end every half hour and lumps of semi-liquid glass being continually withdrawn by bottle-making machines at the other end. The working end is quite



## 1.2.7 THE SCIENCE OF FLAMES AND FURNACES

separate from the melting end, the glass flowing under a bridge between them, and the working end may have separate oil or gas burners for independent temperature control.

Smaller tank furnaces are more frequently fired with rich gas or oil ; in this case only the air is preheated, either by regenerators or metallic or brick recuperators.

The pot-type of glass melting furnace, Fig. (1)22, is somewhat similar to the old-fashioned beehive kiln, the place of the ware in the latter being taken by a ring of pots containing up to 2 tons of glass each. They are usually fired with a built-in producer and air recuperator,\* the gases entering through an "eye" in the centre of the floor and passing out through ports situated in the walls between the pots. A door is placed opposite each pot, the latter often having a spout sticking through the door so that an operator can insert a spoon and withdraw his glass. Alternatively, the pots may be withdrawn and poured (for example, for glass tube manufacture), while optical glass is made by allowing the pot to cool and then breaking it open. The pots are usually made of fireclay, and hence, even though it is high quality, the temperature of the glass cannot exceed  $1300^{\circ}$ – $1400^{\circ}$  C. The pots clearly represent a considerable bottleneck in the heat transfer from flame to glass, but this is to some extent offset by the greater heating surface per unit volume of glass as compared with the tank. These furnaces may be entirely intermittent, all the pots starting from cold at once, or they may be continuous, each pot being refilled as it is emptied. Very clean glass can be made, but the furnace design is restricted by the necessity of allowing for the flow of glass from a broken pot.

**1.2.8. Coal Carbonising Plants.** The primary aim of the coke-oven industry is to make suitable coke for blast furnaces. The gas industry, on the other hand, has the primary aim of supplying a clean gas for domestic and industrial use, with a calorific value varying in different districts from 450 to 550 B.Th.U./ft.<sup>3</sup> (250–306 C.H.U./ft.<sup>3</sup>). Although a number of gas undertakings augment their gas supply by coke-oven gas, obtained from their own coke oven plants or those operated nearby for metallurgical coke manufacture, the two types of carbonising plants are very dissimilar as a result of the differing objectives. A coke oven battery consists of a row of ovens, Fig. (1)23, about 10 ft. high, 40 ft. long, the width

\* In fact the development of brick recuperators has very largely taken place in connection with these furnaces.



varying from 14 to 18 inches,\* the oven walls tapering about 2 inches from the pusher end to the discharge end of the oven. The heating flues are placed between adjacent oven walls, with the corresponding regenerators placed beneath the ovens. The latter are heated either by coke-oven, blast-furnace or producer gas. When using coke-oven gas, the combustion air only is preheated, whereas with the other two fuels both gas and air are preheated. The coking time varies, according to the width of the oven, from 14 to 18 hours, or approximately 1 hour per inch of oven width.† The coal is charged into the ovens from a travelling coal hopper through openings spaced along the top of the oven roof. The carbonised charge is pushed out when it has reached a more or less uniform temperature of  $950^{\circ}$ – $1050^{\circ}$  C., by a horizontal, electrically operated ram, on to the coke wharf or into a coke car. In the former case the hot coke is water-quenched on the wharf and in the latter it is quenched by water sprays from a quenching tower. In some installations the thermal energy of the hot coke is utilised for the production of steam in coke dry-quenching plants of various designs, consisting mainly of a waste gas recirculating system,<sup>1.26</sup> but such systems may result in the combustion of an appreciable fraction of the coke. From a design point of view the thickness of the oven walls must be thin enough to allow a good heat transfer from the heating gases to the charge, but the walls must also be strong enough to withstand the presence of the coke when a charge is pushed. In order to obtain uniform heating of the charge from one end to the other of an oven, careful regulation of the fuel gas supply to each heating flame is necessary, as well as the reversal of the heating gases every 20 or 30 minutes from one half to the other half of an oven wall or from one pair of oven walls to an opposite pair of walls, depending on the design of the coke ovens.

A coke-oven battery consisting of about 60 ovens has a daily coking capacity of about 1000 tons of coal for an average oven width of 18 inches. The gas yield may vary from 11,000 to 12,000

\* The minimum size is fixed by the necessity of the entrance of a bricklayer for repair work.

† This direct proportion is used by some designers, but on theoretical grounds (see Chapter 4) one would expect the necessary heating time to vary as the square of the thickness.

<sup>1.26</sup> SAVAGE, L. H. W., and BRANCKER, A. V.: "The Application of Dry Coke Cooling Plants to Integrated Iron and Steel Works," *J.I.S.I.*, 161, Pt. 2, p. 103, (Feb. 1949).



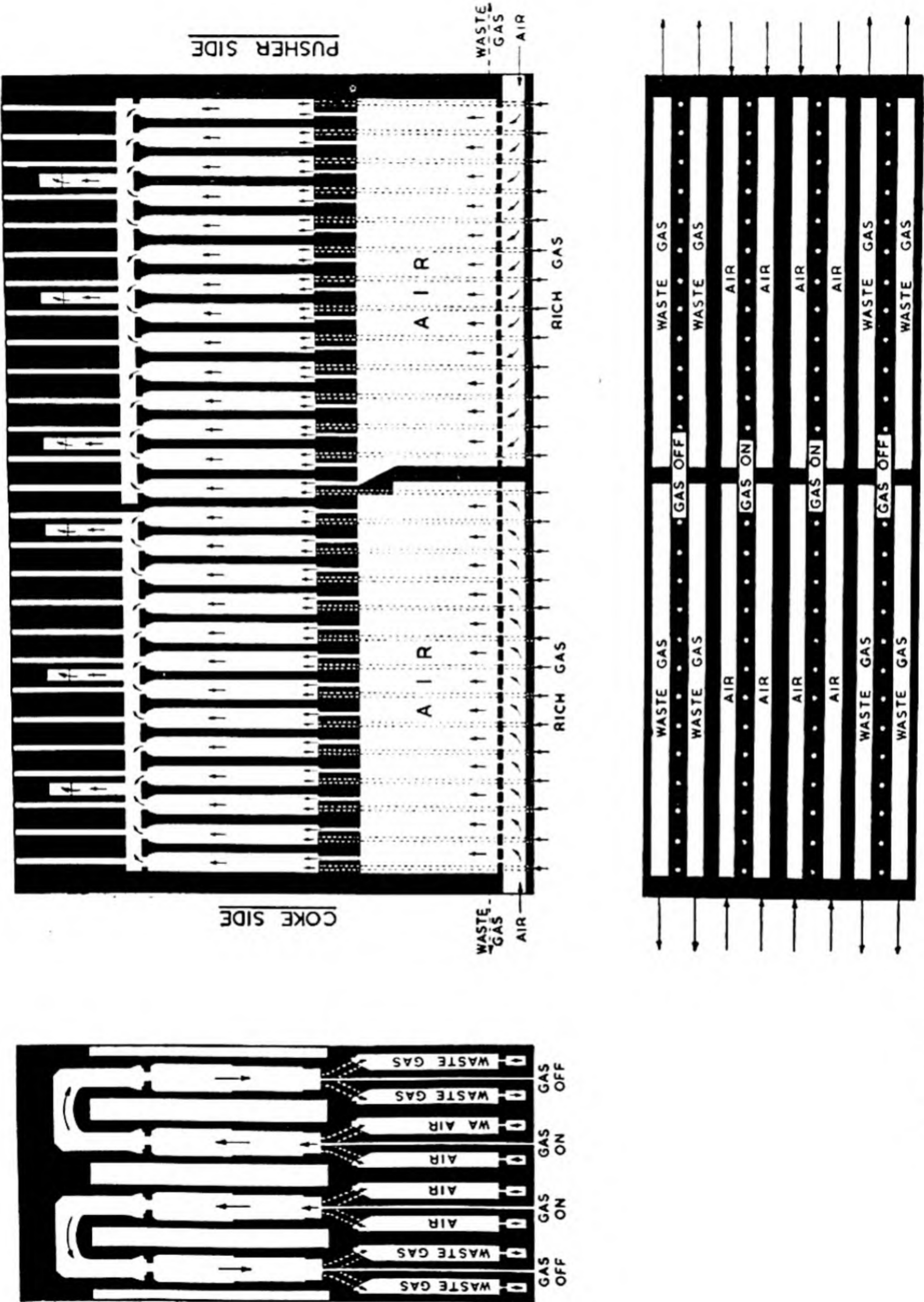


Fig. (1)23 (a). W.-D. Becker Combination Under-Jet Coke Oven (Rich Gas Firing).  
(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company (1920) Ltd.)

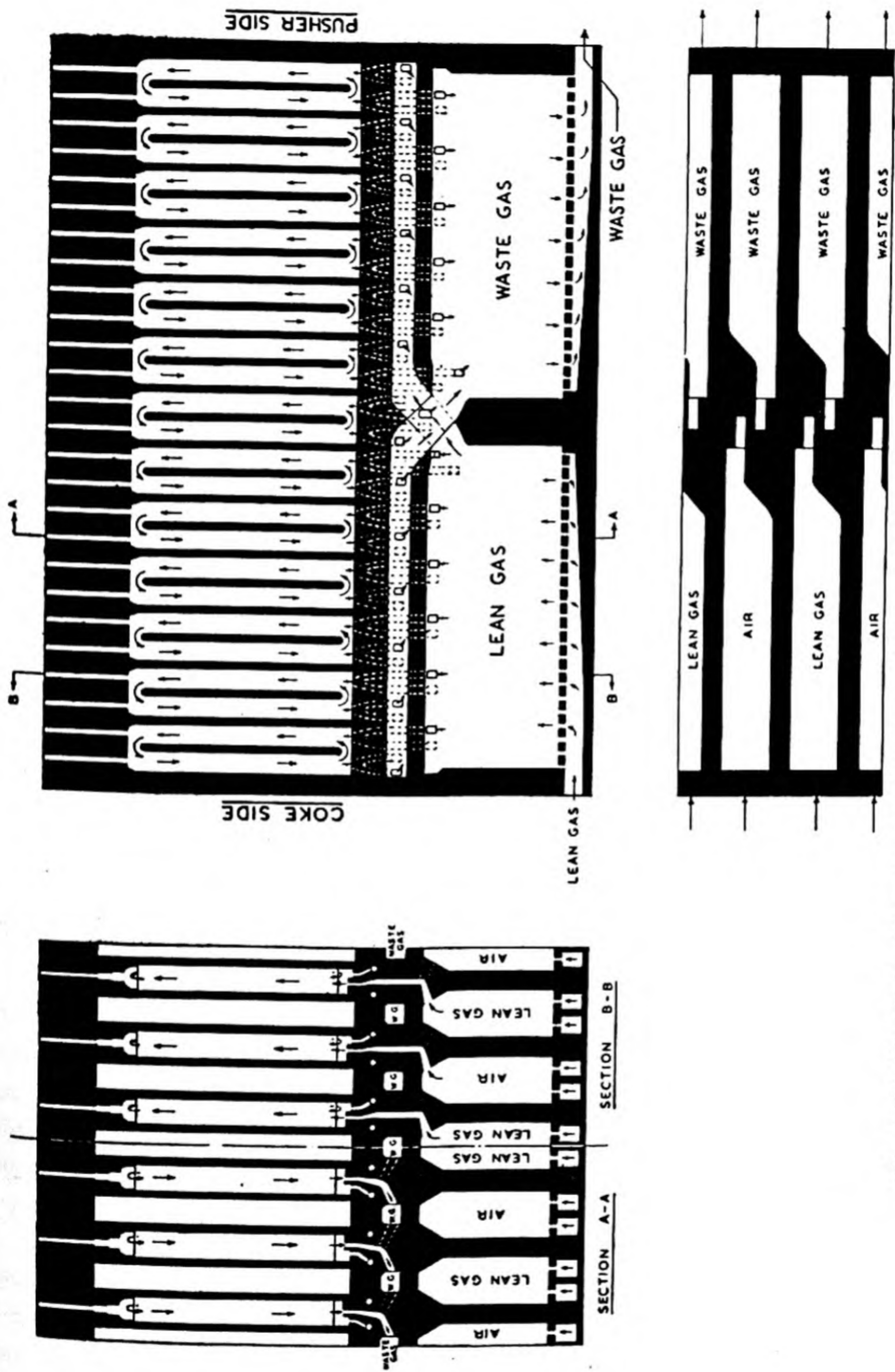


Fig. (1)23 (b). W.-D. Koppers Combination Circulation Coke Oven (Lean Gas Firing),  
(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company (1920) Ltd.)



### 1.2.8 THE SCIENCE OF FLAMES AND FURNACES

ft.<sup>3</sup>/ton, depending on the volatile content of the coal. With coke-oven gas as fuel, about 45% of the gas made is used to heat the ovens, but where gas is valuable for other purposes, as in "integrated" iron and steel works, producer gas made from coal or coke fines is used and both gas and air are regeneratively preheated. The fuel consumption varies from 590 to 670 C.H.U./lb. of coal coked, depending on the temperature of the coke pushed, the type of the fuel gas, the quality of the coal and to a certain extent on the design of the ovens.

In the gas industry, on the other hand, the development from the old type horizontal retort, 10 ft. long and hand-charged, Fig. (1)24, has been towards the following distinct types of carbonising systems :

(1) Horizontal through retort, Fig. (1)25, about 20 ft. long, mechanically charged and discharged. Capacity per retort, approximately 15 tons/24 hours.

(2) Intermittent vertical retort, Fig. (1)26, charge weight 3 to 4 tons, carbonising approximately 7 to 9 tons/24 hours according to the amount of steam passed through the charge to dilute the distilled gas by the formation of water gas.

(3) Continuous vertical retort, Fig. (1)27, continuously fed from an intermediate supply coal hopper, with a continuous coke discharge into a cast-iron hopper at the base of the retort. The output of this type of retort varies from 4 to 10 tons/24 hours. Adjustment of the coke discharge mechanism allows for a considerable variation in the coal throughput. In some cases the coke in the hopper is cooled by steam, which then passes into the retort for the production of water gas.

The process of evolution from the horizontal retort to the vertical retort included the construction of the inclined retort system, which has, however, practically disappeared today in this country. On the Continent the inclined chamber oven, with a daily capacity of 5 to 6 tons, has been widely adopted. The inclined carbonising systems require, however, very delicate regulation to obtain uniform heating.

All these types of carbonising plants are heated by hot raw producer gas from built-in producers or by separate batteries of producers. The combustion air is preheated either in recuperators or, in some types of continuous vertical retorts, in flues surrounding the lower section of the retort where the coke is being cooled and

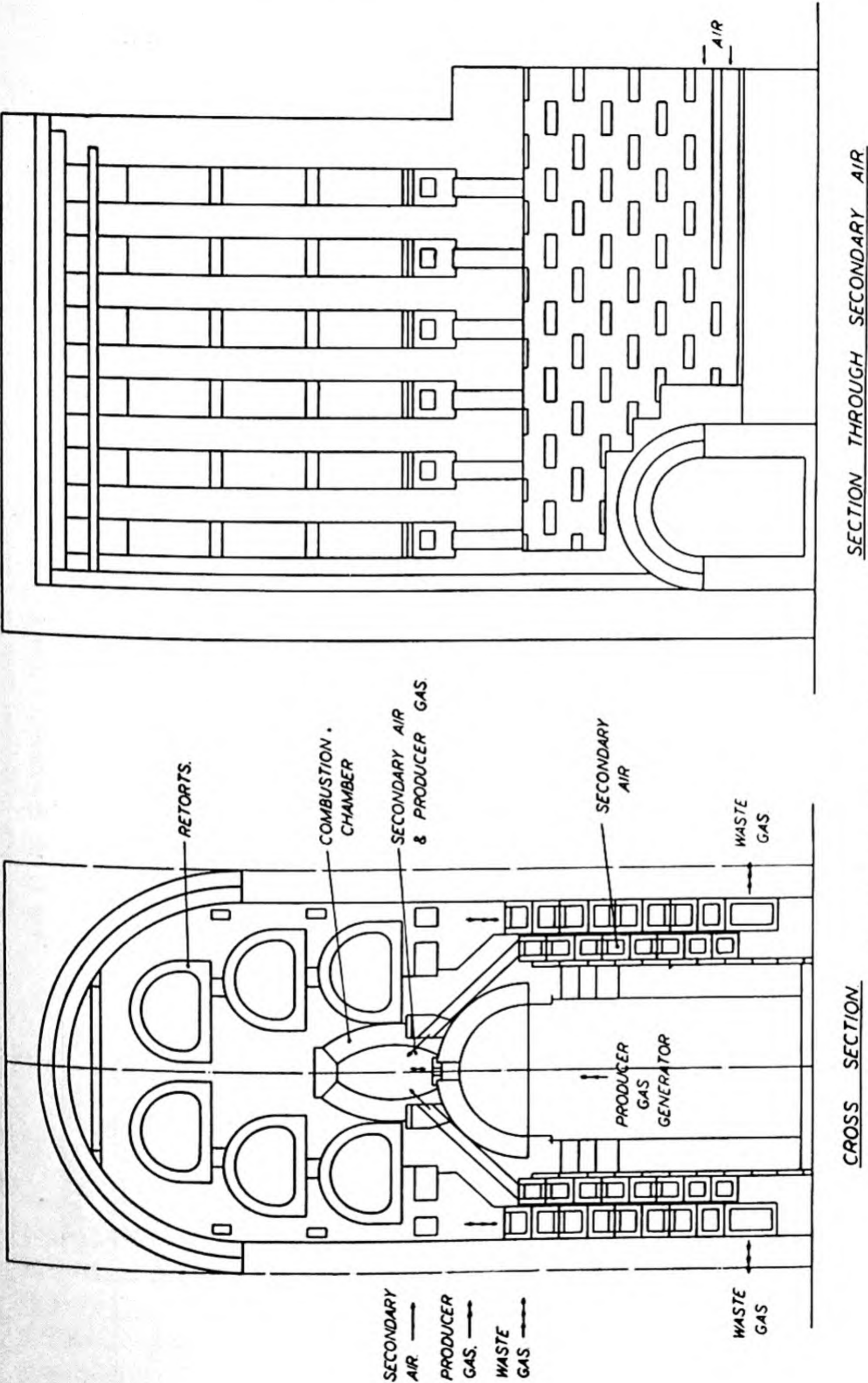


Fig. (1)24. Horizontal Hand-charged Gas Retort.  
(Courtesy of Gibbons Brothers, Limited.)



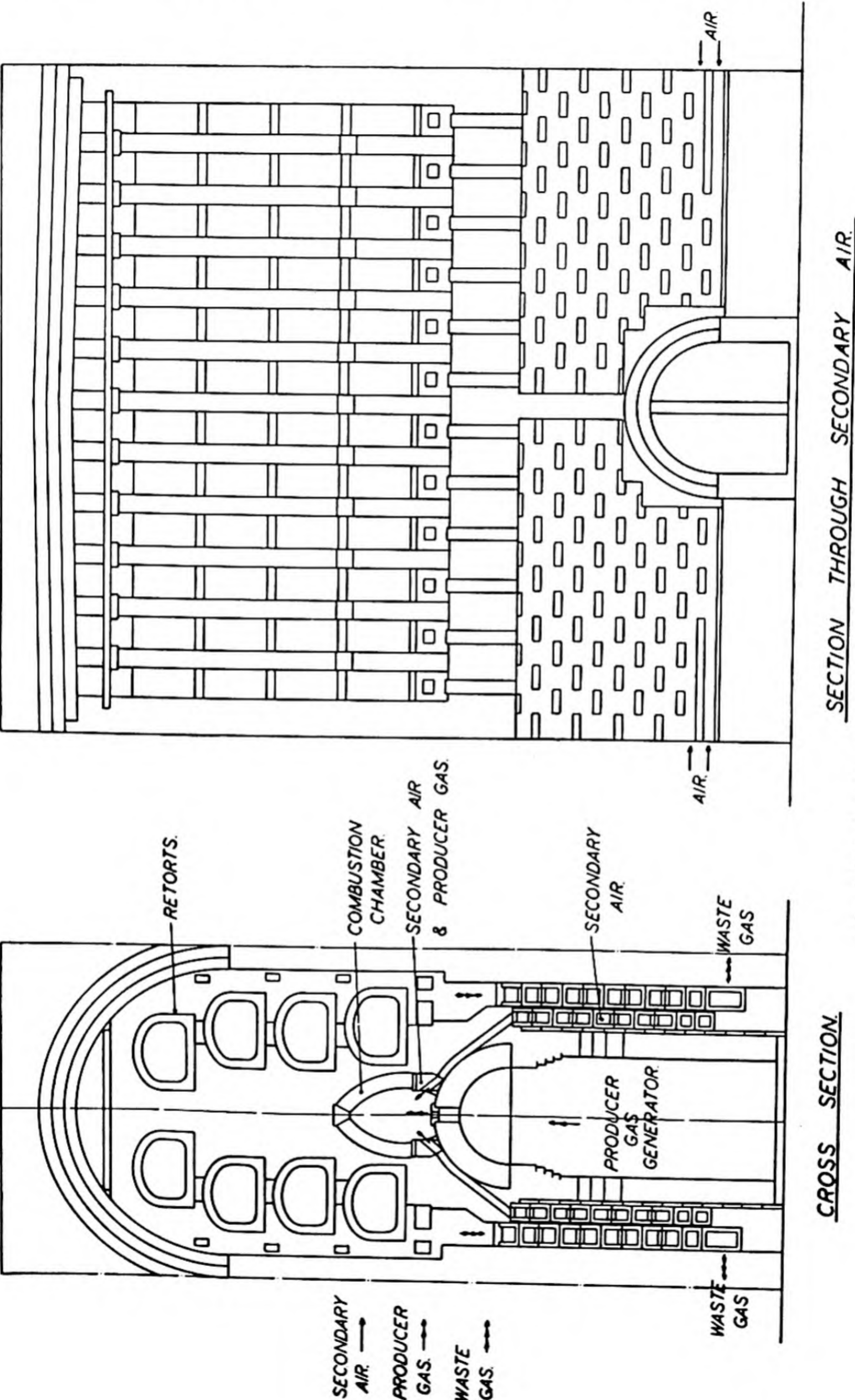


Fig. (1)25. Horizontal Through Gas Retort.  
(Courtesy of Gibbons Brothers, Limited.)

water gas made. This gives a possibility of recovering an appreciable fraction of the sensible heat of the coke, giving as an ultimate objective the use of only the heat necessary for such distillation

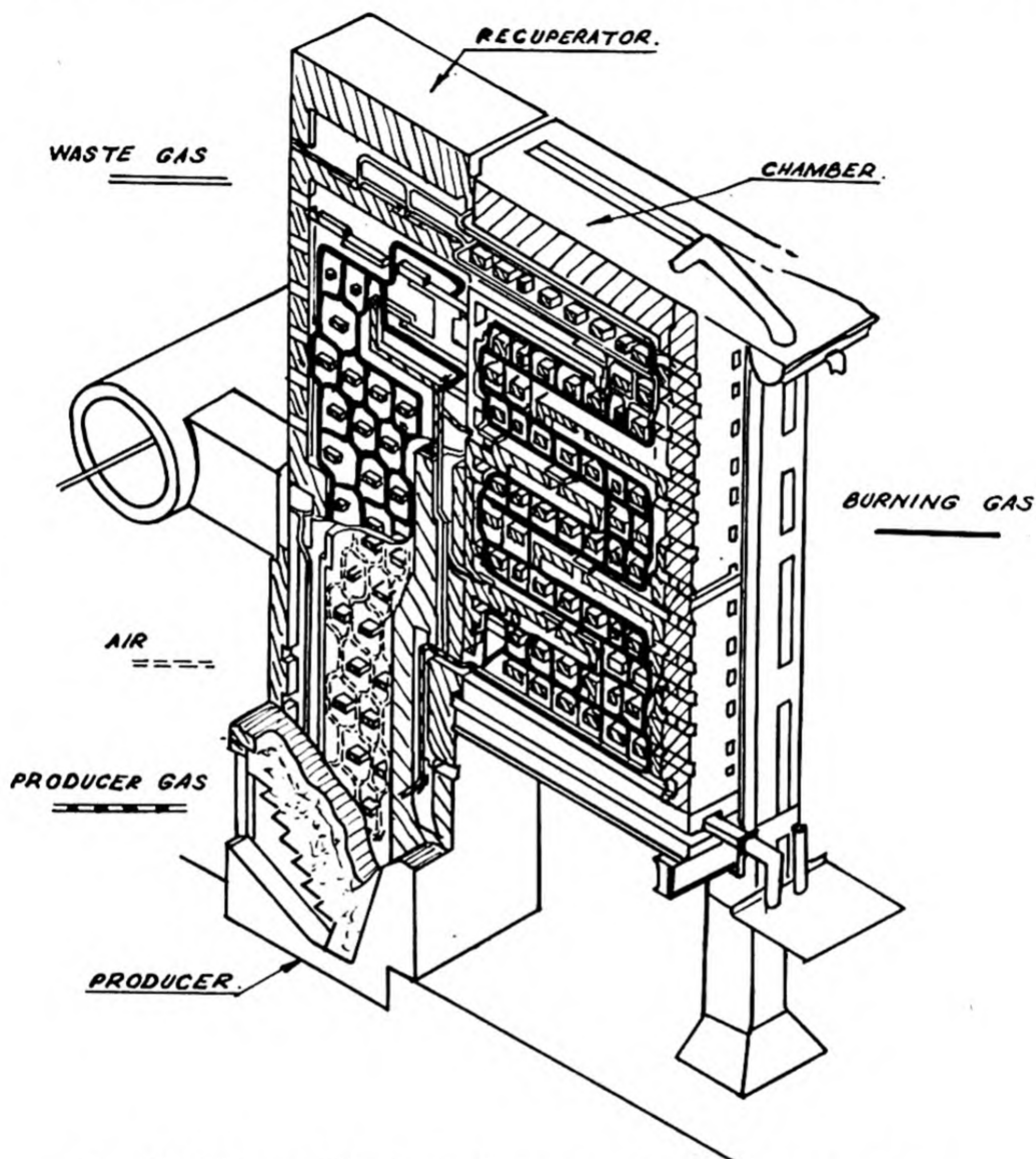


Fig. (1)26. Recuperative Intermittent Vertical Retort—Sectional View.  
(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company (1920), Limited.)

reactions as are endothermic. In present-day continuous retorts the waste gases leave the heating flues at a temperature of approximately  $1000^{\circ}\text{C}$ ., their sensible heat down to about  $230^{\circ}\text{C}$ . being utilised in waste-heat boilers. The resulting steam (about 1500 lb./ton of coal carbonised) is practically sufficient to cover the power



### 1.2.8 THE SCIENCE OF FLAMES AND FURNACES

and steam requirements of the works. The fuel consumption of the intermittent systems is such that approximately 15–16% of the coke made is charged into the producers, but the breeze, which is least saleable, is of course used for this purpose. With the con-

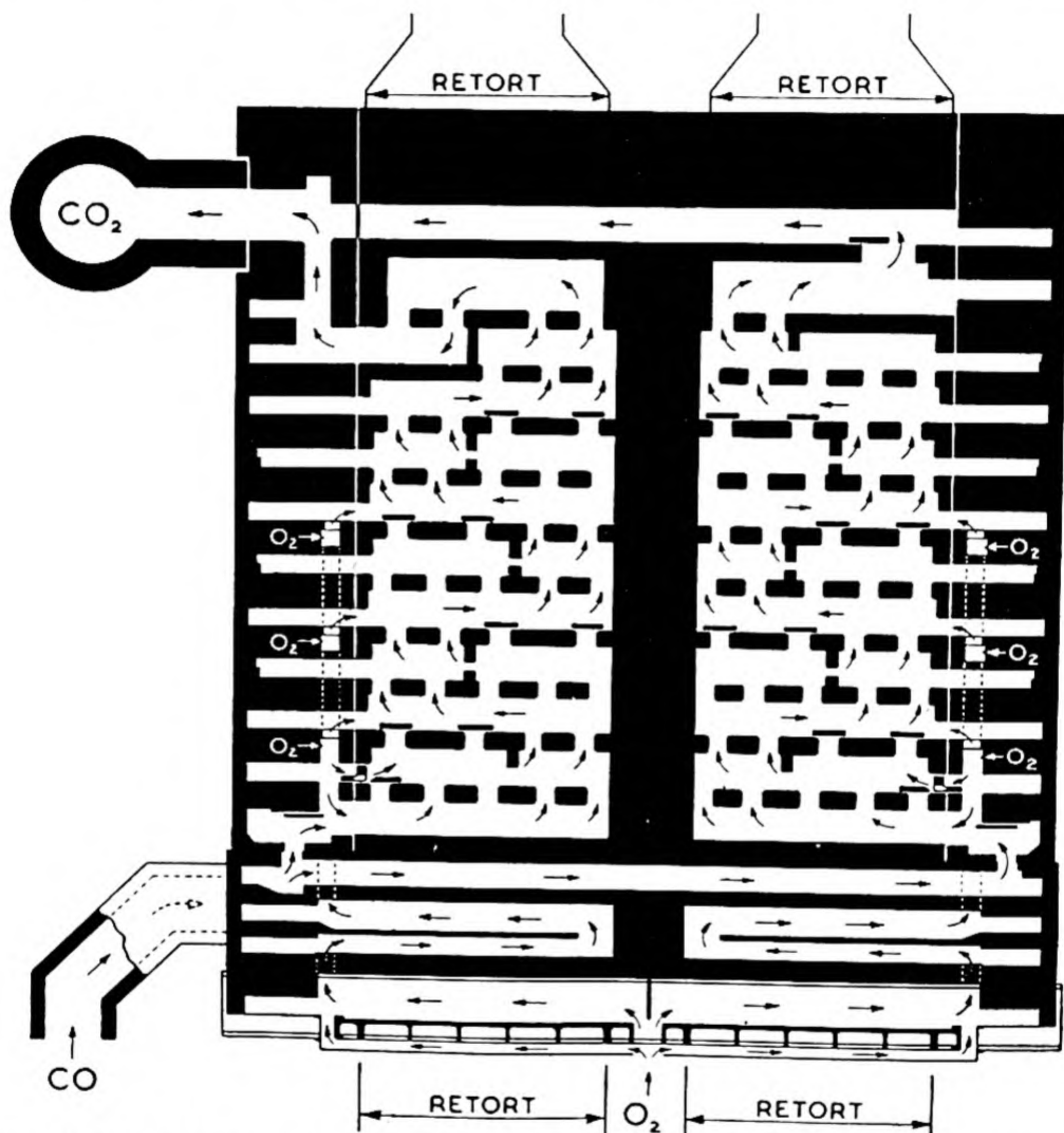


Fig. (1)27 (a). Flow of Heating Gases for Woodall-Duckham Lambent Heated Continuous Vertical Retorts.

(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company (1920), Limited.)

tinuous vertical retort the corresponding coke consumption figures are 10–11% for recuperative heated retorts and about 13–14% for non-recuperative retort settings.

A new type of vertical retort, the *semi-continuous* system, Fig. (1)28, has recently made its appearance. The hot coke is

discharged into a hopper which forms a continuation of the retort proper. Some steam is passed into this hopper for the production of water gas and for cooling the coke. Modern designs of continuous retorts yield well up to 80 therms/ton of coal carbonised with a gas of 450–500 B.Th.U./ft.<sup>3</sup> (250–278 C.H.U./ft.<sup>3</sup>).

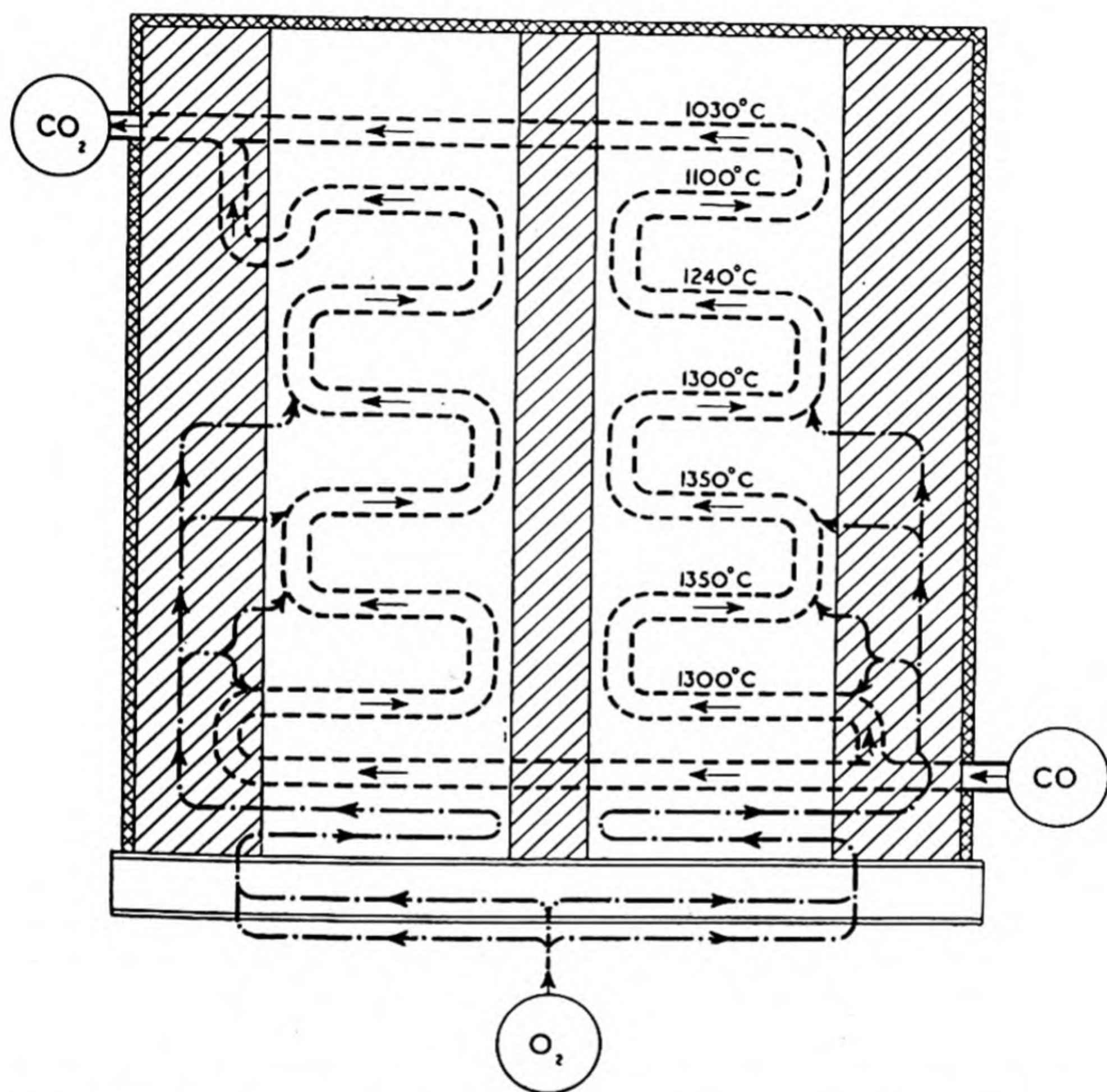


Fig. (1)27 (b) Continuous Vertical Retort with 'Voodall-Duckham System of Lumbent Heating.

(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company (1920), Limited.)

In comparing the systems used in the two carbonising industries it can be said that carbonisation in bulk, as in coke ovens, gives a large output of gas per unit area of ground space, coupled with low costs of labour and maintenance. Continuous retorts are, however, more suitable for gasworks situated in or near a town on account of the smokeless operation, while still maintaining a low cost of



### 1.2.8 THE SCIENCE OF FLAMES AND FURNACES

production per therm coupled with a high yield. Each system is capable of giving very high thermal efficiencies, since both the heat of the gas below the maximum carbonising temperature and of the coke below this temperature can, ideally, be recovered. The use of

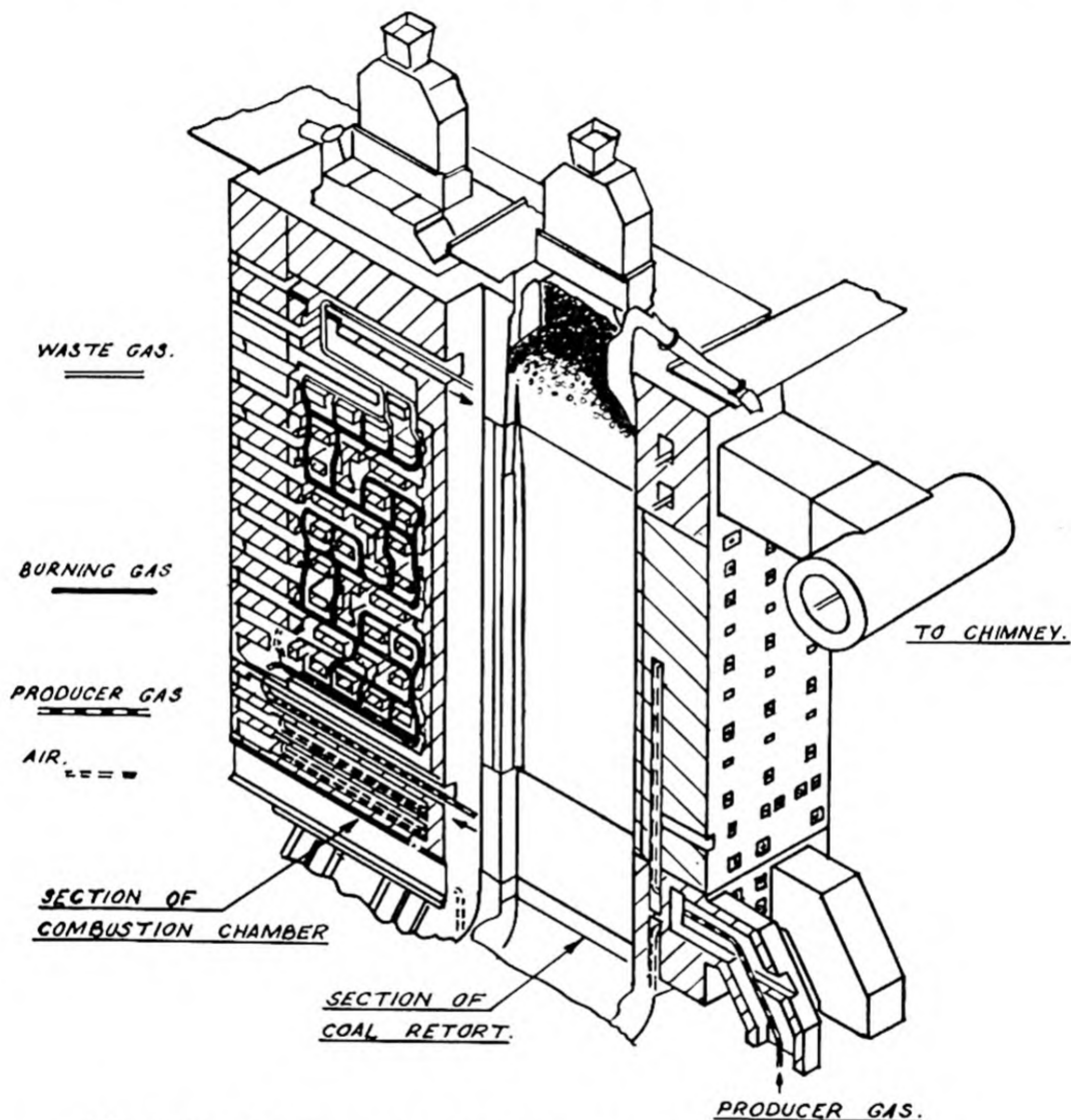


Fig. (1)28. Diagram of Lambent Heated Continuous Vertical Retort.  
(Courtesy of the Woodall-Duckham Vertical Retort and Oven Construction Company  
(1920), Limited.)

the tunnel kiln or of the cyclic-fired kilns of the potteries industries does not seem to have been considered in the carbonising industries, probably because of the low value of unit weight of the material which is heat treated.

Table 1.1 is a summary of the various types of furnaces used in the various industries which have been described in Section 1.2.

TABLE 1.1. GENERAL SURVEY OF THE MAIN INDUSTRIAL PROCESSES USING FURNACES

Industry	Process	Name of Furnace	Type of Furnace
IRON	Iron-smelting	Blast	Direct-fired (coke) shaft.
	Iron-remelting	Cupola Rotary Crucible	Direct-fired (coke) shaft. Coke, oil or gas-fired.
STEEL	Steel-making	Open-hearth Bessemer converter	Regenerative gas or oil-fired bath. Heated by oxidation of metalloids.
	Steel castings	Arc and induction Tropenas or side-blown converter	Electric. Heated by oxidation of metalloids.
	Reheating	Continuous billet-heating Batch-heating	Counterflow continuous. Batch type.
	Ingot-heating	Soaking pit	Batch type.
NON-FERROUS METALS	Smelting	Blast or reverberatory	Shaft or bath-type.
	Re-melting	Reverberatory	Rotary or stationary bath-type.
	Oxidation	Converter	Heat by oxidation of sulphur and iron.
	Castings Heat treatment	Crucible	Oil or gas-fired.
REFRACTORIES	Firing	Beehive kiln	Coal-fired convection heating.
POTTERY AND BRICK	Potteries and refractories	Semi-cylindrical	Gas-fired regenerative.
	Firebricks and building bricks	Semi-continuous kilns	Coal-fired cyclic.
	Firebricks	Tunnel kilns	Straight or circular kiln.
CHEMICAL INDUSTRY	Distillation and concentration	Pot furnace	Direct-heated cast-iron pot.
CEMENT AND LIME	Cement production	Rotary kiln	Pulverised-fuel-fired counterflow rotary kiln.
	Lime-burning	Shaft kiln	Shaft kiln, either gas or direct-fired.
GLASS	Glass-melting	Tank	Gas-fired regenerative bath.
		Pot furnace	Producer or semi-producer or oil-fired recuperative or regenerative muffle.
CARBONISATION	Gas-making	Horizontal Vertical-intermittent Vertical-continuous	Gas-fired muffle. Gas-fired muffle.
	Coke-making	Coke oven	Gas-fired counterflow muffle. Regenerative muffle furnace, gas-fired.



### 1.3. The Industrial Significance of Furnace Processes

Table 1.2 has been compiled in order to give some measure of the value of the furnace output of the various industries of this country.

The figures for this table are taken from the 1935 Census of Production, as this is the last published set of industrial figures. The columns represent: (1) the value in million pounds of the output of the furnaces in the main furnace-using industries; (2) the increase in value of the materials in passing through these industries; (3)–(6) the consumption of coal, coke, oil and electricity for these industries.

Assuming that two-thirds of the net increase in value in these industries is produced by furnace processing, the figure of £160,000,000 as the increase in value of materials in passing through furnaces can be estimated. A figure for the corresponding fuel costs can be obtained as a very rough approximation taking coal at £1 per ton (£22 m.), coke at 30s. (£19.5 m.), fuel oil at 3d. per gallon (£1.5 m.), assuming that one-tenth of the electricity used by these industries is used for furnace heating at  $\frac{1}{4}$ d. per kWh. (£0.5 m.) and that one-third of the gross output of the gas industry is used for industrial fuel (£21.5 m.). This gives a total of £65 m. All these figures would have to be correspondingly increased to represent 1950 values.

### 1.4. Classification of Furnaces and Processes

Apart from the classification by industries given in the preceding section, there are many ways of classifying furnaces each of which has its own particular value. Thus furnaces can be classified by form, i.e. by the way in which the charge is handled. There are *shaft, bath, in and out, pusher, muffle and tunnel furnaces*. The main distinctions of these are shown in Fig. (1)29. Alternatively, furnaces can be classified by the fuel used for heating, the main fuels considered in this monograph being:

- (1) direct coal firing ; ✓
- (2) producer gas made from coal or coke ; ✓
- (3) rich gas (town's gas or coke-oven gas) or mixture of gases ; ✓
- (4) coke mixed in with the charge ; ✓
- (5) oil firing. ✓

From the point of view of the materials of construction, classification into operating temperatures is the most valuable one.

In this monograph, however, the concern is basically with the



TABLE 1.2  
THE INDUSTRIAL SIGNIFICANCE OF FURNACE PROCESSES

	VALUE OF INDUSTRY		FUEL USED			
	Gross Output, £ million	Net Output, £ million	Coal, million tons	Coke, million tons	Fuel Oil, million galls	Electricity, million kWh.
<b>IRON AND STEEL :</b>						
Iron and steel blast furnaces	21.0	4.1	0.34	7.17	—	319
Iron and steel smelting and rolling	101.8	33.6	6.59	0.14	2.75	1043
Iron and steel foundries	39.0	22.9	0.42	0.47	7.85	171
Tinplate	13.9	4.9	0.69	—	0.25	55
Hardware, cutlery, wire, etc.	104.9	51.0	0.99	0.25	25.35	449
	280.6	116.5	9.03	8.03	36.20	2037
<b>NON-FERROUS METALS :</b>						
Copper and brass (and finished brass)	32.8	13.8	0.29	0.11	6.14	160
Aluminium, lead, tin	33.2	9.8	0.38	0.05	4.42	405
Gold, silver, plate, jewellery, watch, clock	41.9	6.3	0.02	—	1.10	23
	107.9	29.9	0.69	0.16	11.66	588
<b>CHEMICAL AND ALLIED :</b>						
Chemical, dyestuffs, drugs	68.8	36.5	3.03	0.22	2.98	1204
Fertiliser, disinfectant, glues	7.3	3.0	0.19	0.02	0.16	31
Petroleum	8.4	3.2	0.21	—	33.38*	48
Coke, by-products and manufactured fuel	17.2	4.4	18.60*	—	0.24	159
Explosives and abrasives	8.1	4.7	0.13	—	0.40	22
	109.8	51.8	3.56	0.24	3.78	1464
<b>CLAY AND BUILDING MATERIALS :</b>						
Brick, fireclay and building materials	43.8	28.4	4.74	0.14	6.12	217
China and earthenware	14.2	9.4	1.10	0.07	0.39	35
Glass	17.2	10.6	0.89	0.05	21.79	163
Cement	9.7	5.6	1.96	0.02	0.42	576
	84.9	54.2	8.69	0.28	28.72	991
<b>GAS UNDERTAKINGS</b>	64.8	39.7	17.89*	4.32	44.20	85
	648.0	292.0	22.00	13.00	124.60	5165

\* These figures are excluded from the fuel consumption totals since they refer to the use of coal and oil as raw materials for processing, not as fuels.



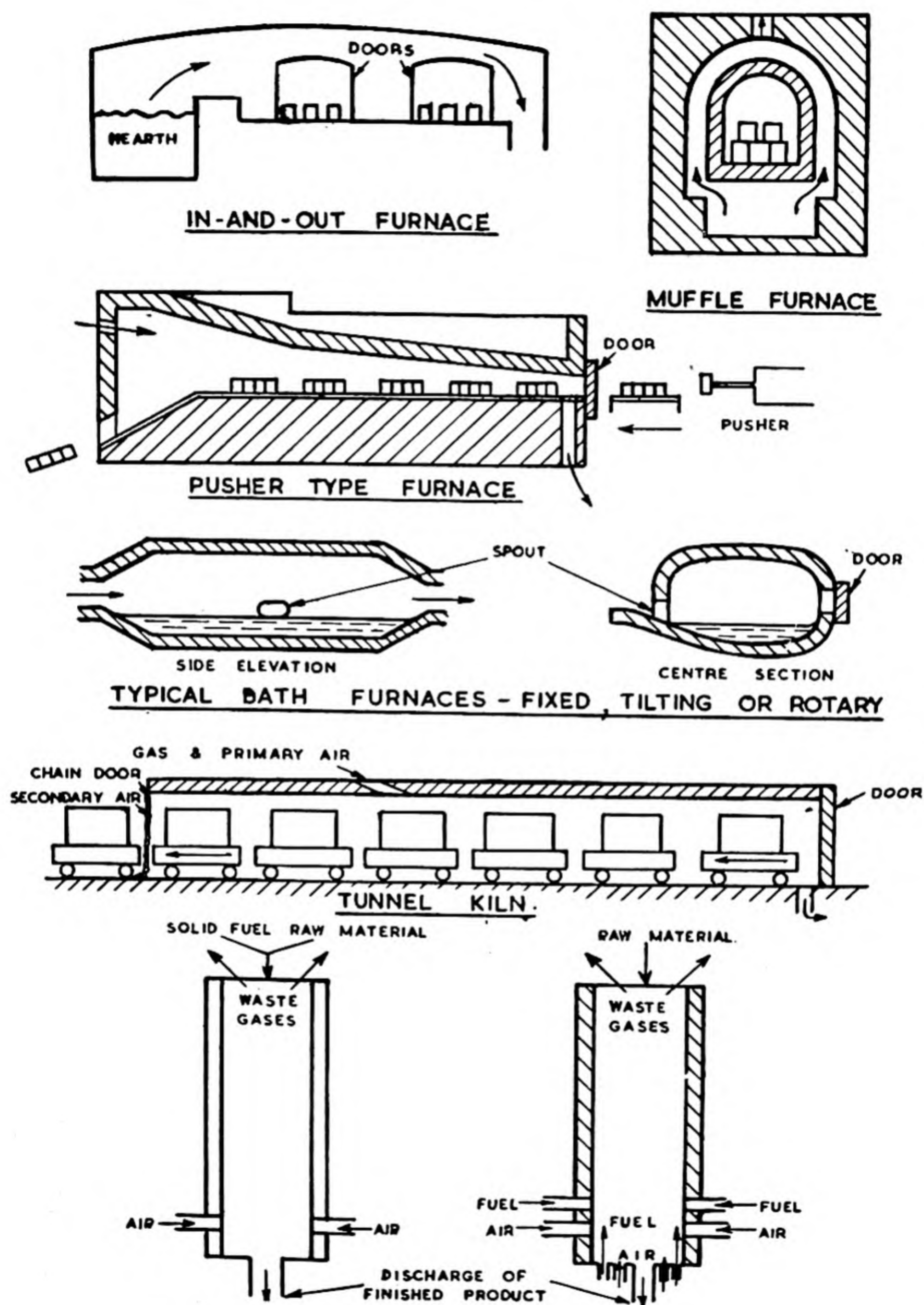


Fig. (1)29. Comparison of Different Types of Furnaces.

processes taking place in furnaces, and for this purpose no single classification is suitable because two furnaces may be alike as far as one process is concerned, but entirely dissimilar from the point of view of another process. For example, the bath type of brass furnace is similar to the open-hearth furnace as far as the process of heat transfer from the flame to the charge is concerned, but entirely dissimilar from the point of view of combustion. For our purpose, therefore, a classification of furnaces does not serve such a useful purpose as a *classification of the processes taking place in the furnaces*.

It is suggested that the basic processes are those covered by the following branches of knowledge which are accordingly required for furnace design and operation.

(a) *The Thermodynamics of Heating Processes*. This includes : (i) the consequences of the first law of thermodynamics, that is, the conception that heat is one particular form of energy with constant conversion factors relative to the chemical energy of a fuel, to electrical energy or to mechanical work ; (ii) the consequences of the second law of thermodynamics, that is, the conception that these different forms of energy have different measures of availability ("virtue"), which means that energy of one form can only be converted into a higher grade form at the expense of a certain fraction of itself ; (iii) the consequences which these conceptions give as regards the relative values of heat in different parts of the furnace, such as the undesirability of delayed combustion and of large temperature differences across which heat flows, and hence the desirability of high heat transfer coefficients in furnaces (heat transfer laws).

(b) *The Laws of Energy Release by Combustion of Fuels*. The amounts of energy that can be released by combustion, and the factors governing the speed and completeness of combustion.

(c) *The Laws of Flow of Hot Gases*. These cover the factors governing the quantity of gas and air that pass through a furnace system, the laws governing the sub-division of gases flowing from a single channel into separate ones, and the laws governing the velocity distribution, persistence of jets, the magnitude of eddies, stratification, and of stagnant pockets in wide channels. These are of importance because they affect local heat transfer, refractory erosion and rates of combustion.

(d) *The Science of Construction* of the walls or "skin" of a furnace, which includes the properties of various materials at high



## 1.4 THE SCIENCE OF FLAMES AND FURNACES

temperatures, taking into account the relation between the amount of fuel which is necessary to produce them in usable form from materials present in nature, and the amount of fuel which their use will save: the designs of structure suitable to carry the loading with the relatively poor physical properties available at high temperatures, and the designs of internal structures for separating parts of the system, viz. muffles and heat exchange tubes.

The next five chapters of this monograph deal with these four subjects, the basic laws of thermodynamics (Chapter 2) being dealt with before combustion (Chapter 3), while heat transfer, although a thermodynamic subject, is considered separately in Chapter 4, the concluding chapters being a note on the right use of the principles of science on a complicated practical problem of this type, and a brief survey of the main requirements in the way of further knowledge.

## 2. THE THERMODYNAMICS OF FURNACE HEATING

### 2.1. The Application of the First Law of Thermodynamics to Furnaces

**2.1.1. Statement of the Law.** The first law of thermodynamics—the law of conservation of energy—is based on the experimental fact that various things, such as a lump of coal, a ray of sunshine or a charged accumulator, have in common the power to produce heat in their surroundings. This power is called energy, and it is found that there is a constant conversion factor when each of these forms of energy is converted entirely into heat. The factors by which energy can be expressed as heat units are given in Table 2.1.

TABLE 2.1. ENERGY/HEAT CONVERSION FACTORS

	C.H.U.*
Mechanical energy 1 ft./lb.	0·000714
Electrical energy { 1 watt/sec. 1 kW./hr.	0·000527 1900
Chemical energy { 1 lb. of carbon burnt to CO <sub>2</sub> at 25° C. 1 lb. of hydrogen burnt to H <sub>2</sub> O at 25° C. Latent heat of 1 lb. of steam at 100° C.	7860 34,200 583

The first law summarises all such experiments and may be stated: *the change of internal energy of a body in any process is the difference between the total energy entering it and the total energy leaving it in the process.* The forms of energy which are particularly important in the science of furnaces are: (1) the chemical energy of unburnt and partially burnt fuels and of the material being heated where this undergoes reactions; and (2) the sensible and latent heat of combustion gases, heated materials and furnace construction materials.

The application of the law can be illustrated by the case of a gas-air mixture supplied to a furnace. Combustion in the mixture

\* In this book the Centigrade temperature scale is used throughout and the ft./lb. mechanical system ; the unit of heat is thus the Centigrade Heat Unit (C.H.U.), the heat required to raise 1 lb. of water through 1° C. A more complete set of conversion factors is given at the end of the book.



### 2.1.1. THE SCIENCE OF FLAMES AND FURNACES

reduces the chemical energy of the gases but increases their sensible heat, and the application of the law to the gases flowing through a certain region gives the equation:

$$\begin{aligned} &\text{Reduction in chemical energy of gases in going from } A \text{ to } B \\ &\quad - \text{increase in sensible heat of gases} \\ &= \text{total heat transferred by the gases to surroundings in going from} \\ &\quad A \text{ to } B. \end{aligned}$$

Similarly, the law may be applied to a certain section of a furnace in which case it gives the relation

$$\begin{aligned} &\text{Rate of heat input to charge} + \text{total heat losses from the furnace} \\ &\text{section} \\ &= (\text{chemical and sensible energy of gases entering section per} \\ &\quad \text{unit time}) \\ &\quad - (\text{chemical and sensible energy of gases leaving per unit time}). \end{aligned}$$

**2.1.2. The Heat Balance.** The first law of thermodynamics is used *implicitly* in all heating calculations, for example, the calculation of the expected performance of a furnace, but its particular *explicit* application is to the construction of the heat balance. A heat balance is a flow sheet for the various forms of energy entering and leaving a furnace or part of a furnace.

Various ways of expressing the different quantities of energy concerned in the heat balance are possible. In furnaces operating in the steady state (continuous furnaces) the heat balance is expressed in energy units per unit time (hour) or per unit weight of product, whereas in batch-type furnaces it is better to consider the total number of energy units flowing at each point in a specified period of the heating process, such as one complete cast on an open-hearth furnace. In the discussion below we shall consider only the construction of a heat balance of the first type, except where special differences arise in the construction of the second type of balance. The units in which the energy is expressed may be simply heat units (C.H.U.) or lb. of standard coal, cubic feet of standard gas or gallons of standard oil. It is clearly necessary to reduce actual measurements to the equivalent measurements on a fuel of a specified calorific value, since, for example, an increase in the ash content of the coal supplied would be unfairly counted against the process if the weight of coal used was given in both cases without correction.

Having determined the input energy, it is in some cases convenient



to use this as a scale for the diagram, expressing it as 100 and all other energies as percentages of this value. This is useful in that it gives a direct representation of the thermal efficiency of the process as the part of the energy stream flowing to the product, but it is not appropriate when the aim of the heat balance is to assess the fuel savings of a change of operation or design of the furnace. In the latter case it is recommended to use as unit 1 lb. of standard fuel (e.g. coal of gross C.V. 12,600 B.Th.U. per lb.) per 100 lb. of processed product, as this shows directly the magnitude and cause of fuel savings when the heat balances for two furnaces are compared. A heat balance can apply either to the whole of the furnace or can be built up to relate separately to the different parts of the furnace system, e.g. in an open-hearth system the bath, regenerators and gas-producer can be separated. In the one case it is necessary to know the heat input and output in various forms from the furnace as a whole, in the other case these quantities must be known for each part separately. In either case, however, the heat entering and leaving takes one of the following forms :

(a) *Chemical energy of the fuel or partially burnt gases.* This is the product of the calorific value (C.H.U./lb. or C.H.U./ $n$  ft.<sup>3</sup>) and the quantity of the fuel or gases (lb./hr. or  $n$  ft.<sup>3</sup>/hr.).\* The calorific value of a combustible gas in C.H.U./ $n$  ft.<sup>3</sup> or of a solid or liquid fuel in C.H.U./lb. may be calculated from a knowledge of the chemical compounds present in the fuel and the heats of reaction of these compounds with oxygen (for some typical reaction energies see Section 3.1.2) or from measurements of the calorific value of the fuel made in a calorimeter. In general, for furnace calculations the former method is used for simpler fuels and the latter for more complex fuels or gases of unknown composition. In order to make an approximate calculation of a furnace the exact fuel for which is unknown, tables of the average properties of typical fuels, such as those given in *Technical Data on Fuel*,† are convenient. These refer respectively to gaseous liquid and solid fuels. The Seyler chart (*Technical Data on Fuel*, p. 394) is also very convenient for obtaining the combustion characteristics of coals whose hydrogen and carbon contents (dry-ash-free basis) are known.

(b) *Sensible and latent heat of hot gases.* These can be calculated from the quantity of these gases ( $n$  ft.<sup>3</sup>) and their temperatures,

\* The symbol  $n$  ft.<sup>3</sup> is used to denote cubic feet measured at N.T.P.

† Edited H. M. Spiers. 5th Edn. 1950. Publ. British National Cttee. World Power Conference.



## 2.1.2 THE SCIENCE OF FLAMES AND FURNACES

using the total (sensible) heat content tables in *Technical Data on Fuel*, p. 161. Alternatively, the Rosin-Fehling I.T. diagram, Fig. (2)1, may be used to obtain the sensible heat (C.H.U./ $n$  ft.<sup>3</sup>) of the combustion products from usual fuels with various ratios of combustion gas to air at various temperatures. This diagram allows for dissociation effects by including an abnormal specific heat, a technique which has also been used by Ribaud.<sup>2.0</sup> In calculating a proposed furnace, the temperatures of the gases leaving each section

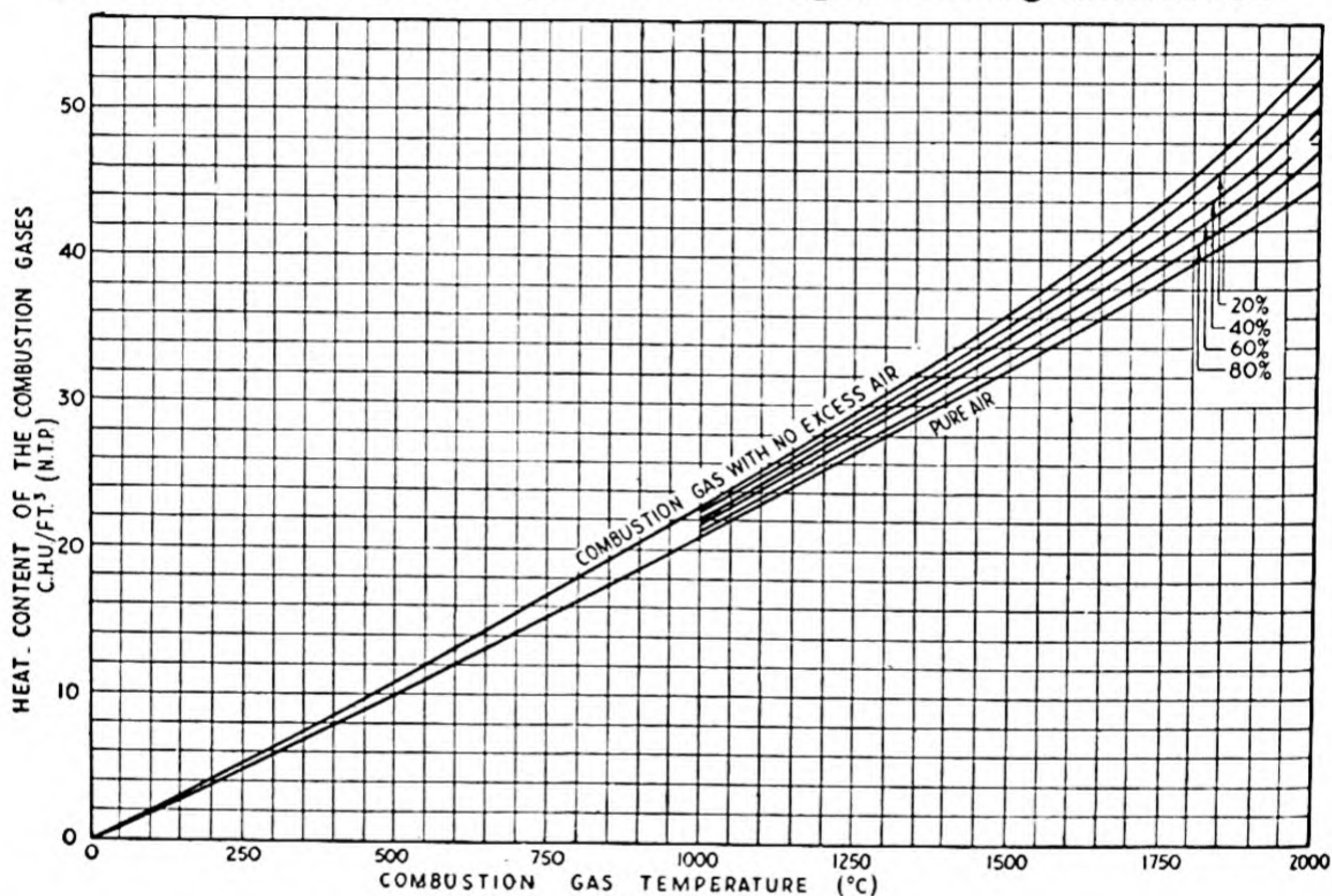


Fig. (2)1. Rosin-Fehling I.T. Diagram relating the Heat Content of Combustion Gases from all normal fuels to their Temperature.

(Courtesy of Dr. P. O. Rosin and Dr. H. R. Fehling.)

Figures attached to the intermediate curves denote the percentage of air in the mixture of combustion gas and air.

have to be assumed, either based on a standard figure, e.g. 100° C. above the temperature of the charge in that section, or by making it equal to the prevailing furnace temperature. In the case of the construction of a heat balance for an existing furnace, however, measurement of the outgoing gas temperature by some means (see Section 4.1) is preferable for the calculation of sensible heat terms. Alternatively, the figure required for the heat balance is given

<sup>2.0</sup> RIBAUD, G. M.: "Sur la notion de chaleur spécifique apparente," *Chaleur et Industrie*, p. 235 (1937).



directly by measuring the sensible heat content of the gases per unit volume (see also Section 4.1).

(c) *Heat losses by radiation and convection* from the particular part of the furnace system. For a proposed furnace these should be calculated according to the method given in 2.6. For furnaces which are already in existence they may be calculated from measured values of the external wall temperature, using the curves attached to 2.6, or measured directly by means of the heat-flow gauges of Blackie<sup>2.1</sup> and others.<sup>2.2</sup> Where the furnace structure is in an unsteady state and changes temperature during the period of the heat balance, allowance must be made for change of the heat stored in it. This can be done by estimating a mean temperature for the brickwork at the beginning and end of the period and multiplying the heat content per lb. corresponding to the difference of these temperatures by the weight of the brickwork. A better way is, however, to construct the heat balance for the total period of the heating and cooling cycle so that the brickwork is at the same temperature at the beginning and end. For furnaces which cool down over the week-end, a considerably lower thermal efficiency (higher fuel consumption per unit of product) will be obtained if the balance is constructed to include the whole weekly cycle than if it is made on the basis of a single charge or day. The calculation of over-all fuel costs is more accurately made on the former basis. It is even desirable to make allowance, preferably as a separate item, for the heat used in initially bringing the furnace up to temperature, since it is the year's fuel bill which really counts on a process, and a furnace which gives a low fuel consumption for a short period and then has to be cooled and rebuilt may give a much higher fuel consumption over a year than one of slightly worse performance on an individual cycle but which runs for three times as long without shutting down.

(d) *Heat given up to the charge.* This is the sum of the heat absorbed by chemical reactions (such as the reduction of oxides) in the charge, and the increase of sensible and latent heat of the charge. The heat absorbed by some typical chemical reactions is given in Table 2.2: figures from this table multiplied by the hourly reacting weight of the particular compound in the charge which reacts, give the total heat absorbed by such reactions. Where the reactions

<sup>2.1</sup> *J. Scientific Instruments*, 7, p. 7 (1930); 10, p. 364 (1933).

<sup>2.2</sup> BRADLEY, C. B.: "Industrial Heating," *A.S.M.E.*, 12, pp. 1318-20 (Aug. 1945).



## 2.1.2 THE SCIENCE OF FLAMES AND FURNACES

give out heat the figures must be subtracted from, instead of added to, the heat absorbed by the charge in increasing its temperature.\*

TABLE 2.2. HEAT ABSORPTION BY BATCH REACTIONS

FeO	→ Fe + $\frac{1}{2}$ O <sub>2</sub>	Absorbs	890 C.H.U./lb. of FeO	
Fe <sub>2</sub> O <sub>3</sub>	→ 2Fe + $\frac{3}{2}$ O <sub>2</sub>	„	1240	„ Fe <sub>2</sub> O <sub>3</sub>
Fe <sub>3</sub> O <sub>4</sub>	→ 3Fe + 2O <sub>2</sub>	„	1150	„ Fe <sub>3</sub> O <sub>4</sub>
SiO <sub>2</sub>	→ Si + O <sub>2</sub>	„	3160	„ SiO <sub>2</sub>
CaCO <sub>3</sub>	→ CaO + CO <sub>2</sub>	„	427	„ CaCO <sub>3</sub>
MgCO <sub>3</sub>	→ MgO + CO <sub>2</sub>	„	306	„ MgCO <sub>3</sub>
FeCO <sub>3</sub>	→ FeO + CO <sub>2</sub>	„	121	„ FeCO <sub>3</sub>

In this table the fuels such as H<sub>2</sub>, C or CO which may cause the reduction occurring in the first four reactions are not included, so that any quantity of them introduced with the charge can be credited separately as fuel, using the figures of Section 3.1.2 appropriate to whether it is partially or totally oxidised.

The increase of sensible and latent heat of the charge is obtained from the temperature rise using tables of total heat content or specific heat. (For metals, see *Technical Data*, p. 164, where the figures in cal./gm. are the same as C.H.U./lb. ; for refractories, *Technical Data*, p. 171 ; cement clinker, p. 171 ; slags, p. 172 ; and iron oxides, p. 173.)

Where the charge is passing continuously through the furnace the heat given to the charge is the product of the heat absorbed (chemical and sensible) by unit weight in going through the section concerned, and the weight passing per hour. Where the furnace is of the batch type the heat balance will vary from moment to moment of the heating process and may be constructed for a certain part of the heating cycle (e.g. the melting period of the open-hearth furnace) or for the whole heating cycle. Since heat balances are usually constructed with the aim of improving fuel economy it is preferable to include the whole process, including the fuel used while there is nothing in the furnace.

The heat losses from the “skin” of the furnace are frequently estimated by the difference of the other items, i.e. by applying the first law of thermodynamics. This procedure is very unsatisfactory, however, as in this case the first law cannot be used to

\* A typical example is the Bessemer converter, where molten iron at 1300°–1400° C. is converted to molten steel at over 1600° C. without the use of any fuel, and blowing it with cold air, by the heat of oxidation of the carbon and other elements in the iron.



provide a check on the other figures unless the heat losses appear as negative, when the law indicates that some part of the data is seriously wrong ! When the heat losses are estimated separately, as described in 4.4, it is easy to see whether the heat output balances the heat input with the accuracy that would be expected. If a sufficiently good agreement is not obtained, each item has to be critically re-examined, and this process will often bring to light factors of considerable interest. Such an agreement is highly desirable, since the practice of leaving the residue as "radiation and unaccounted for," while just permissible with boilers when the main item (the heat in the steam) is accurately known, is quite inappropriate for furnaces.

**2.1.3. The Sankey Diagram.** The construction of a furnace heat

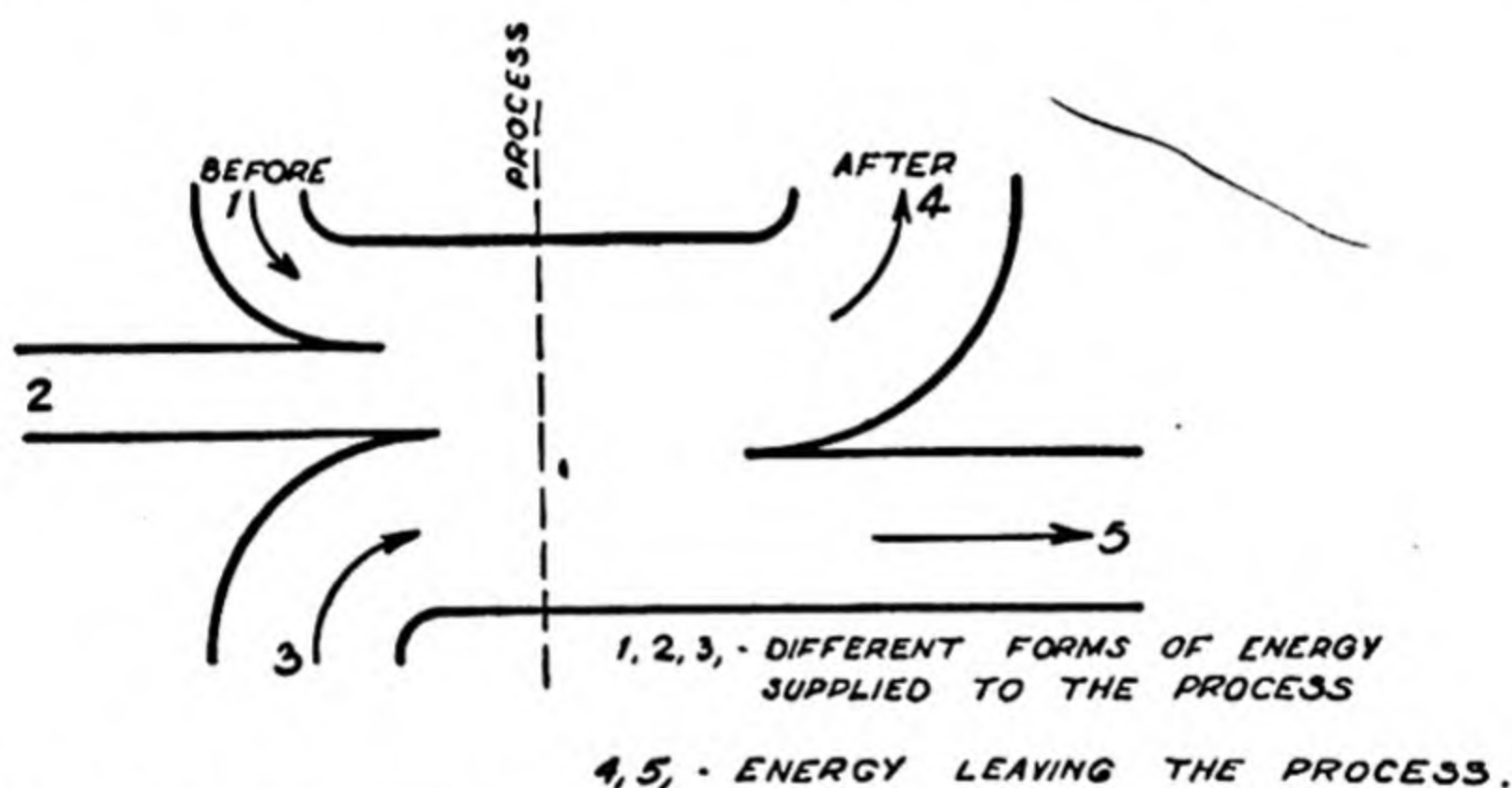


Fig. (2)2. General Diagram illustrating an energy conversion process.

balance is greatly assisted and made more useful if the results are presented in graphical form by means of the Sankey Diagram. This consists of a two-dimensional map of the parts of the system on which is superimposed a stream representing the flow of energy. The width of this stream at a given point is proportional to the number of units of energy flowing past that point expressed in one of the ways discussed at the beginning of Section 2.1.2.

The conservation law then implies that any process which is shown by tributaries leaving or entering the stream is characterised by the fact that the sum of all the parts before this process is equal to the sum of the parts after it. This is shown diagrammatically in Fig. (2)2, where

$$(1) + (2) + (3) = (4) + (5).$$

The continuous recuperative billet-heating furnace fired with



### 2.1.3 THE SCIENCE OF FLAMES AND FURNACES

coal in a deep fuel bed provides a useful illustration of the construction of a Sankey Diagram. The first part of the process is the partial combustion of coal in the fuel bed to give a hot gas containing a certain percentage of combustible ( $\text{CO}$ ,  $\text{H}_2$  and gaseous hydrocarbons). To draw the corresponding part of the energy flow stream we require to know :

- (1) the amount of coal fed to the producer per unit time, and
- (2) the calorific value of this coal ;
- (3) the weight and temperature of the steam supplied under the bed per unit time (if any) ;
- (4) the temperature of the gases leaving the producer and their composition (from which the sensible and chemical energies of these gases can be calculated), and
- (5) the inside or outside surface temperature of the brickwork around the fuel bed and its area (from which the heat losses can be calculated).

Where it is not convenient to measure the temperature (or sensible heat content) and chemical energy of the gases above the fuel bed,\* the heat balance of the bed cannot be separated from that of the furnace. In such a case the energy at this point is determined by working back from the rest of the system, and the first law can only provide a check on the accuracy of the calculations for the system as a whole (e.g. it cannot be used to separate the heat losses of the furnace from those of the fuel bed).

Suppose the gas is then burnt with preheated secondary air : this air brings back from the flue gases an amount of energy which can be calculated from measurement of the secondary air temperature together with direct measurement of the volume, or else calculation of the volume from the gas analyses before and after secondary combustion. This secondary combustion process usually takes place simultaneously with heat transfer to the charge and walls and hence it is not necessary in the diagram to separate it from the heat transfer. The heat transferred to the charge is calculated from the weight treated per unit time and the temperature rise, the heat

\* Although they make the heat balance of much greater value, such measurements are very rarely made because the apparatus (suction pyrometer and water-cooled gas probes) is elaborate. In the case, however, of those gas producers which are quite separate from the furnace, a greater proportion of the total energy of the gas is in chemical form and the gas is cooler, so that measurements of its energy leaving the producer become both easier and more essential.



losses are calculated (usually by assuming a figure for the inside wall temperature, see Section 4.4 below) and the heat leaving the furnace in the form of sensible heat of gases is obtained by measurement of the gas temperature or sensible heat/unit volume.\*

By measuring the composition of the gases entering and of those leaving the furnace and the fuel consumption, the total secondary air can be calculated. In practice much of the secondary air comes in through furnace doors and brickwork so that it is not preheated in the recuperator. To construct an accurate heat balance it is therefore necessary to obtain a measure of this "false air"—this can be done by metering the air to the recuperator, so that the air leakage is obtained as the difference between the metered quantity

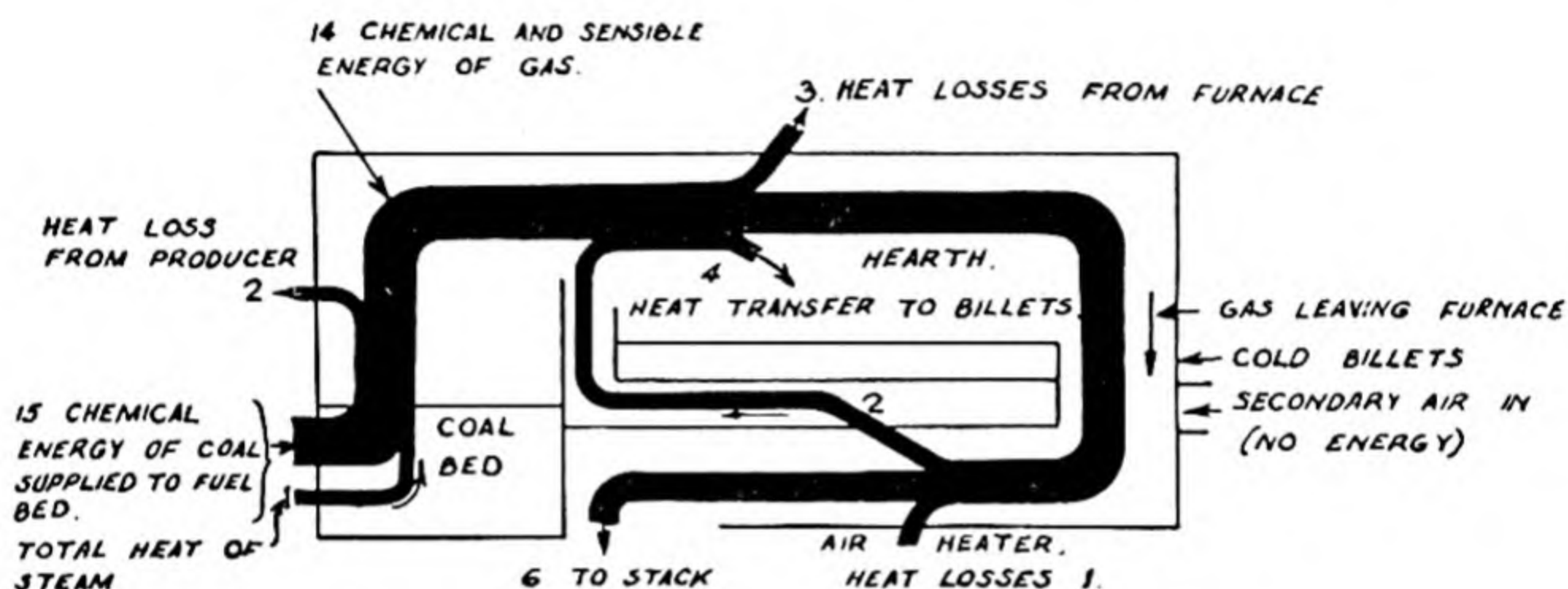


Fig. (2)3. Sankey Diagram for Billet-heating Furnace.

and the calculated total. In just the same way, a balance can be struck for the recuperator, using measurements of the temperature of the air and gas before and after passing through it and estimating the amount of air infiltrated into the waste gases by analyses of samples taken before and after.

In this way the diagram of Fig. (2)3 was constructed. Similar diagrams can be constructed for any process in which energy is converted from one form to another, ranging from an electric motor to a bunsen burner ; another example is provided by the cement kiln for which the diagram of Fig. (2)4 has been calculated. The energy input is obtained from the weight of standard coal, calculated from the measured actual coal used to fire 100 lb. of clinker. The

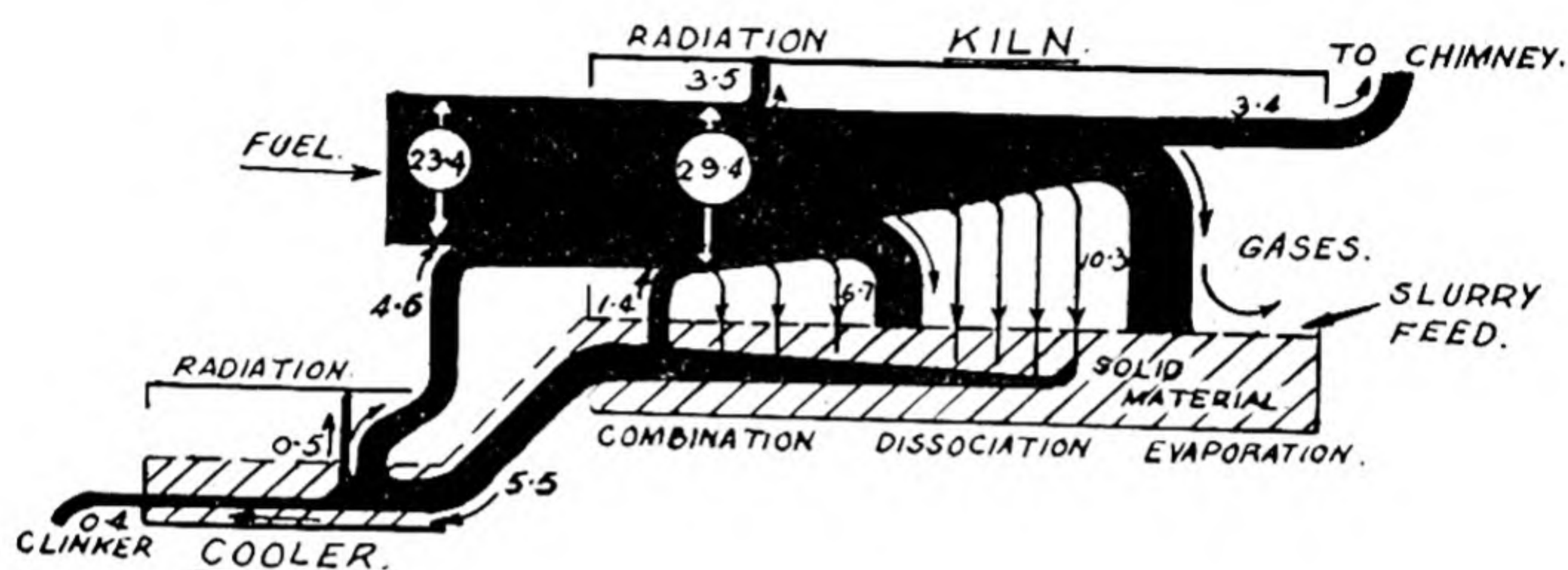
\* Here again, if this measurement is impracticable the heat balance of the furnace chamber cannot be separated from that of the recuperator, but this time the measurement is of such great importance that the heat balance has very little value in showing the causes of inefficiency if it cannot be made.



### 2.1.3 THE SCIENCE OF FLAMES AND FURNACES

heat capacity of the charge and the heat absorbed by the cement-forming reactions are calculated from the known changes of weight and composition, the reaction energies and the weight of water in the slurry; the sensible heat of the gases leaving the kiln is calculated from their measured temperature and composition, and the heat carried by the clinker into the air heater and that brought back

GOOD.



BAD.

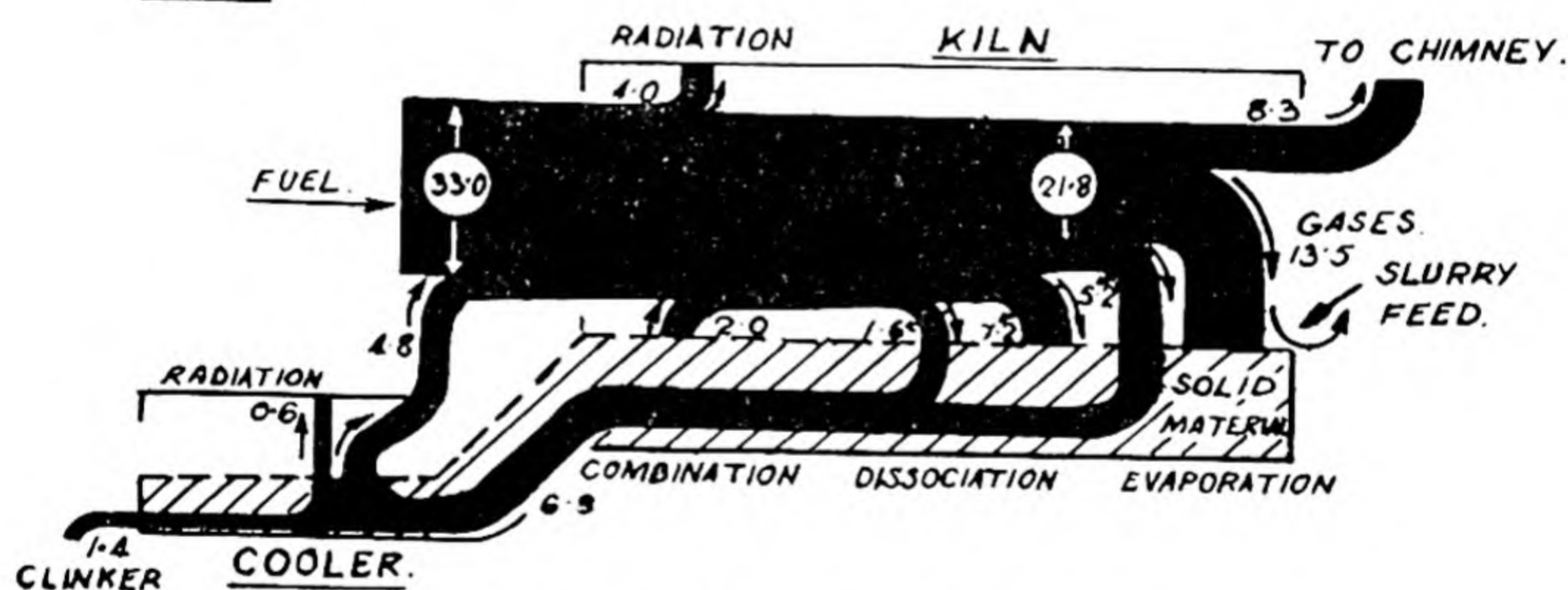
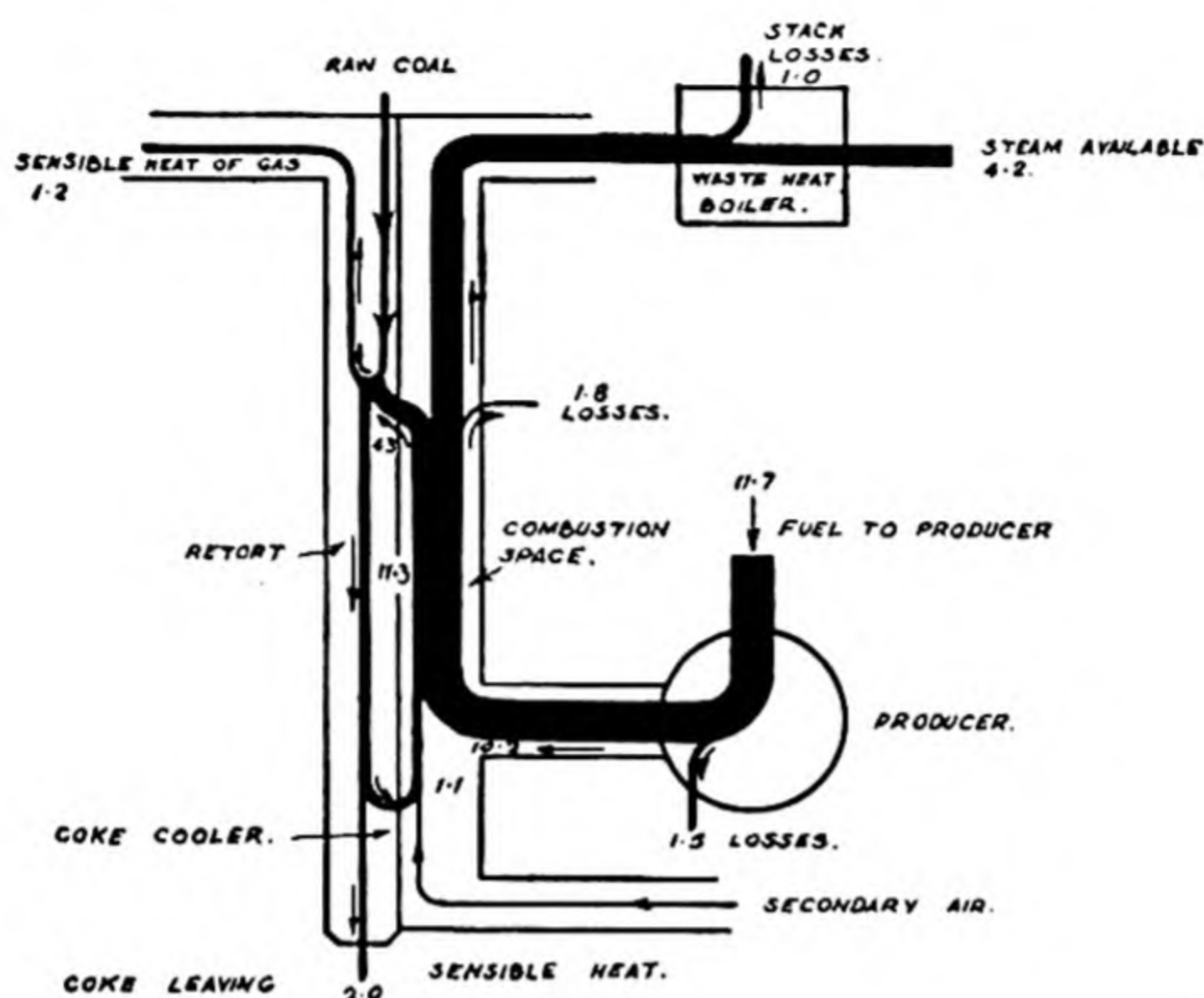


Fig. (2)4. Sankey Diagram for Cement Kilns.

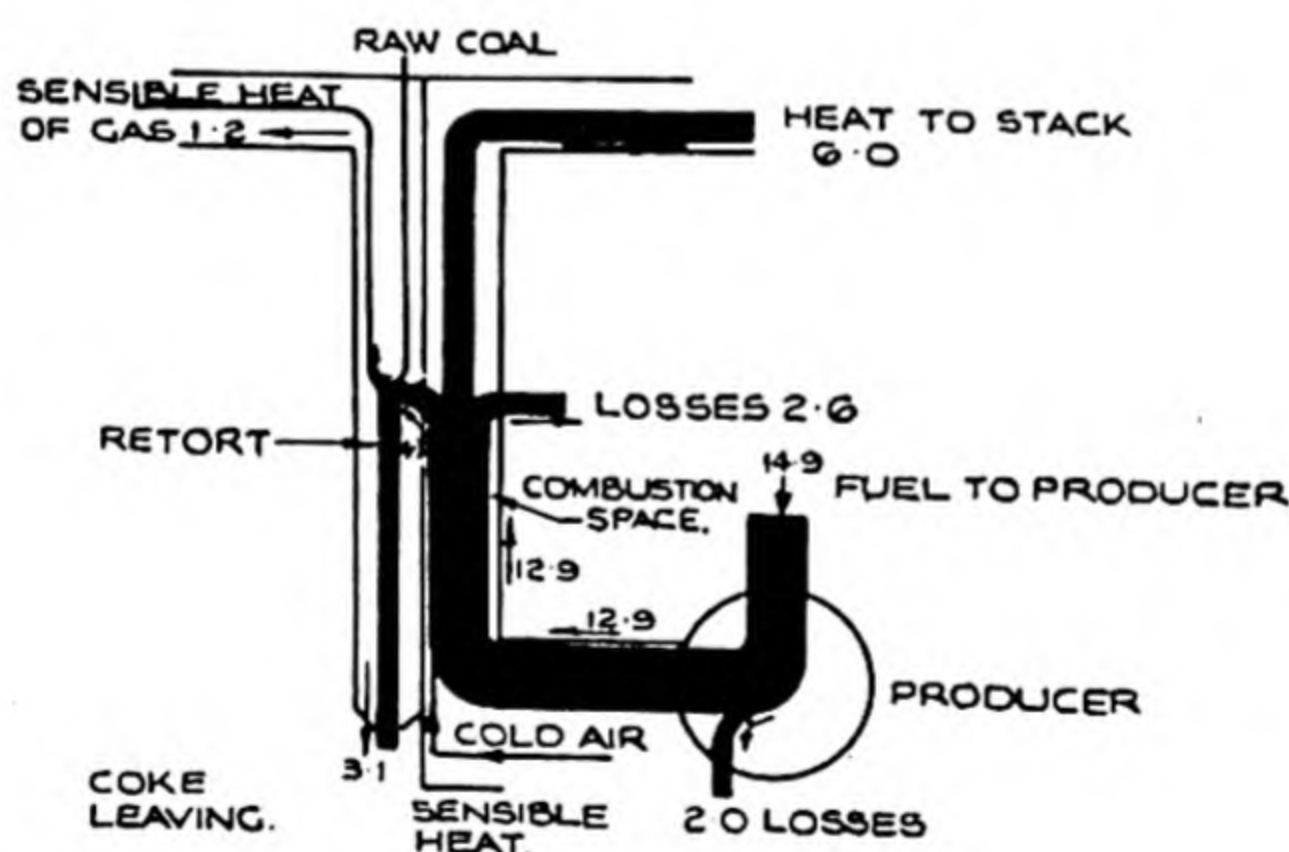
by the preheated air are calculated from temperatures of clinker and air measured in the ducting connecting the air heater (clinker cooler) and the main rotary kiln.

A simpler way of representing the results of an over-all heat balance for a furnace system is shown in Fig. (2)5, which has been constructed for a coke oven. This method is useful where no attempt is made to split up the balance into parts, i.e. to separate different points where the same kind of loss occurs.

**2.1.4. The Meaning of Efficiency.** So far the construction of the heat balance has been straightforward and heat balances of this type can be carried to any required degree of accuracy by refinement



(a) Good practice.



(b) Bad practice.

Fig. (2)5. Sankey Diagram for Coke-Oven Retorts. (All figures refer to pounds of standard coal per 100 lb. of coal carbonised.)

of the measurements. On the other hand, when such a heat balance is used to define a furnace *efficiency* there is considerable room for disagreement, which lies in deciding what are the true requirements of the process in thermal units. This arises from the



#### 2.1.4 THE SCIENCE OF FLAMES AND FURNACES

fact that there are distinct stages in development of fuel consciousness in an industry. The most primitive stage is that in which a supply of energy is used to produce a selected result without any regard to the amount of energy necessary for that result. The next stage is an attempt to increase the efficiency to as large a fraction of 100% as possible, where efficiency is defined as the ratio of the energy required to obtain the selected result to the energy actually used. The third and most advanced stage is the investigation of ways of getting what is really needed without attaining the selected result at all ; in other words, this stage involves showing that the problem has been wrongly formulated. This may seem a paradox, but has in fact been achieved by certain industries.

The three stages may be illustrated by the example of gas-making. The first step is the realisation that if coal is heated to a certain temperature, gas and coke are obtained. At this stage coal is heated in any kind of a vessel without regard to the amount of fuel burnt to heat it. This roughly corresponds to the state of affairs when gas was used mainly for illumination purposes. Next comes the calculation that coke and gas have such and such specific heats and hence that their production requires a certain minimum quantity of energy to heat them, and provide any heat which may be required for decomposition reactions. All efforts are then directed towards reducing the energy down to this quantity. Finally, usually before the efficiency at this stage has reached anywhere near 100%, it is realised that gas and coke require a certain temperature in the process of their manufacture, but that, once made, they are wanted for use in the cold state, and hence that the minimum over-all energy essential for the process is only that required for actual reactions of decomposition, which is practically zero. In other words, it requires  $x$  B.Th.U. to heat 1 lb. of coal to  $1000^{\circ}$  C. and split it up into gas and coke, and between 98 and 102%\* of these  $x$  B.Th.U. are theoretically recoverable by cooling the gas and coke, which has to be done in any case before the latter can be used. This stage is illustrated by the development of continuous retorts in which the coke leaving the hottest carbonising zone is partially cooled by preheating the secondary air for combustion by heat transfer outwards through the walls of the lowest part of the retort ; the gas leaving the hottest zone is cooled to a large degree by counterflow with the incoming coal, and the greater part of the heat of the combustion gases leaving it is used for steam raising.

\* Some coals undergo exothermic reactions on heating.



This same argument applies to many industries, the only outstanding exception being those in which strongly endothermic reactions take place, such as in the manufacture of calcium carbide from lime and carbon, and those where a form of high-grade energy is required for power production, as in the power-plant boiler. Regarding a boiler by itself, i.e. apart from the steam-using appliance, it has to supply steam at a certain temperature and pressure, and thus obviously cannot do this with less than the number of heat units corresponding to the internal energy of the steam. Similarly, if the steam is used to make another form of energy, e.g. in a turbine, the limit is set by this energy conversion. When, however, the boiler is looked at in conjunction with the steam-using appliance, then if the steam is used for process work the processed materials are usually required in the cold state and hence the true minimum energy requirements are nearly zero.

Another very typical example is in the refractories industry. Here the firing process is required to produce very little endothermic reaction and the bulk of the energy is used purely for sensible heat, and, as the product is required cold, is recoverable. The three stages of development in this case are as follows :

First, the products are burnt in a directly fired kiln and all the heat in the gases leaving the kiln passes up the stack. Secondly, an attempt is made to reduce the heat consumption to the minimum required by the heat capacity of the charge, by using all the heat of the gases below the charge temperature by means of waste-heat boilers or by heating the charge in counterflow with the heating gases. In the third stage the charge as it cools is made to preheat incoming combustion air as well as using counterflow heating. In this way the fuel consumption could ideally be reduced to that required to make good the heat losses through the wall and that required for endothermic reactions. The technical development of these kilns is then a matter of improving the heat transfer in the attempt to cool perfectly the outgoing bricks at one end and the outgoing gases at the other.

Some industries have largely reached the third of these stages. An example is the sugar industry, which uses far less energy than the latent heat of the water evaporated. Others, such as the gas industry and the refractories industry, are spread out across all three stages. Others again, such as the steel industry, which calls the sensible heat of the molten steel its theoretical heat requirement



## 2.1.4 THE SCIENCE OF FLAMES AND FURNACES

and in many works remelts cold pig iron, are almost entirely in the second stage. Whether it will be possible to utilise the heat of the cooling ingots and thus reduce the fuel consumption to zero (for steelmaking is an exothermic process) remains to be visualised in the future. It must, therefore, be concluded that the meaning of the word "efficiency" is different according to the stage in which a process finds itself. For an industry in the second stage, efficiency has one definite meaning. Here it is the ratio between (1) the total amount of heat required to bring the raw material *up to the maximum temperature of the process* and carry out all the desired reactions, and (2) the total energy of all the fuel actually used. When the third stage is reached efficiency again has a simple meaning. Here it is net amount of heat required to bring the cold raw materials to the *cold finished state* divided by the total energy in the fuel supplied.

It is very important to emphasise this distinction between different types of efficiency, since, even on the first law of thermodynamics alone, it is possible to find an "efficiency" defined in the way appropriate to the second stage greater than 100% when the method is applied to an industry in the third stage.

The second law of thermodynamics indicates the possibility of obtaining "efficiencies" which, even according to the definition appropriate to the third industrial stage, would exceed 100%. These have not yet, however, been achieved, with one exception,<sup>2.3</sup> the heating of rooms by reversed refrigerators, where an amount of heat equal already to about three times the equivalent of the electrical energy used can be pumped from the cooler surroundings into a room, and the theoretical amount is even greater when the required room temperature is not too far above that of the surroundings.

## 2.2. The Application of the Second Law of Thermodynamics to Furnace Systems

**2.2.1. The Quality of Energy.** The first law of thermodynamics says, as we have seen, that the total energy which enters any conversion process whatsoever is equal to that which leaves the same process. What this law leaves unsaid is that different forms of energy have different qualities and that the *quality* of the energy concerned is always lowered in actual processes. The second law of thermodynamics leads to a numerical evaluation of this lowering

<sup>2.3</sup> WALL, T. F. : "The Heat Pump," *Engineering*, p. 221 (19 March 1943).



of quality. The law is stated in some such form as the following : "It is not possible for a self-acting machine working in a cyclical process unaided by any external agency, to make heat pass from one body to another at a higher temperature."<sup>2.4</sup> It follows immediately from this statement of the law that qualities of energy can be compared by the fact that energy in form *A* can be said to have higher quality than that of form *B* if it can be made to flow entirely from form *A* to form *B* in a system in which no other energy is involved. Thus coal can be burnt, electric power consumed in a resistance, or falling weights brought to rest, to give in each case heat energy, or heat can be allowed to flow from any body to any colder body. In this way a scale of energy quality can be set up at the summit of which are put electrical energy, potential energy of a raised weight and kinetic energy of a moving body, while bodies at successively lower temperatures down to that of the surroundings form the steps of the scale.

The second law of thermodynamics can then be stated by saying that energy can only flow unaided *down* the scale of quality. This simple idea of energy quality can be applied to furnaces by calculating the fraction of a form of energy which is available at any temperature.\* Thus suppose a certain mass *m* lb. of perfect gas has constant specific heat *c* (C.H.U./per lb.). If this gas is at any temperature  $T_M$  it will have an amount of heat energy  $c(T_M - T_O)$  C.H.U. available above room temperature  $T_O$ . The fraction of this amount of energy which is available to heat a body which is already at a temperature  $T_A$  is

$$\frac{T_M - T_A}{T_M - T_O}$$

2.4 ROBERTS, J. K. : *Heat and Thermodynamics* (Blackie), p. 250.

\* In the treatment of this chapter the second law of thermodynamics is applied in terms of the concept of virtue rather than that of entropy, because it is felt to be more readily grasped in the former way as far as the application to furnaces is concerned, for two reasons. Thus entropy measures disorder and the entropy of a system in which irreversible changes occur always increases, whereas it is easier to grasp a quantity which, like the energy of a badly insulated furnace, leaks away rather than one which increases. Secondly, virtue is attached to the energy as it is handed on from one object to another, whereas entropy belongs to the object, so that the second law can be stated in terms of entropy in two ways : one, that the entropy of an ideal object (e.g. a perfect gas) is a function only of the conditions of state of that body, the other that the entropy of a closed system remains constant in a reversible process. The virtue formulation is that the virtue of the energy concerned remains constant in a reversible process and decreases in an irreversible one.



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This available heat fraction, which can be called  $f$ , is plotted against  $T_A$  as a percentage. In the case of a gas of constant specific heat a straight line would be obtained such as that shown in Fig. (2)6a. This straight line would cut the line for zero available energy at  $T_M$ . The figures for the actual combustion gases of various fuels fall below the straight line at higher temperatures owing to the increase in the specific heat and to dissociation. A set of such curves is shown in Fig. (2)6a-f.

Such diagrams are very useful in furnace design. For example, in a counterflow furnace such as a cement kiln or a blast furnace, a corresponding curve can be plotted for the charge if the total heat of the charge can be expressed on the same scale, i.e. that in which the total heat of the gas is given as 100%. Since the transfer of each unit of energy from the gases to the charge corresponds to a certain position in the furnace, the diagram then gives a kind of map of what is taking place. The energy scale has a one-to-one correspondence to positions along the furnace, although the relation is not of course necessarily uniform. The second law of thermodynamics then simply says that the curve for the heated material can at no point lie outside the curve for the heating material. Moreover, the temperature difference at any value of  $f$  is the actual temperature difference available for transfer of heat at this point, a figure which will be needed for the heat transfer calculations discussed in Section 4.3. Such diagrams can also be constructed on a basis of energy per unit of product, and this method has recently proved of great value in comparing different effects such as those of preheat and oxygen enrichment in the blast furnace. They make clear, however, the point that the so-called thermodynamics can really only be used for static calculations, since the effect of variation of output rate, which is a time effect, cannot be calculated from such a diagram at all, but requires the assessment of how much the gas temperature curve must lie outside the charge temperature curve, which depends on heat transfer and chemical reaction rates, which are dealt with in Chapters 3 and 4 respectively.

Similarly, in a batch-type furnace, where the gases must leave the furnace chamber at a temperature exceeding that of the final batch, the diagram shows the maximum percentage of the gas heat energy which can be used directly in the furnace chamber at this temperature. It thus enables the improvement in efficiency due to preheat or to a lowering of the exit temperature to be rapidly assessed and made clear.



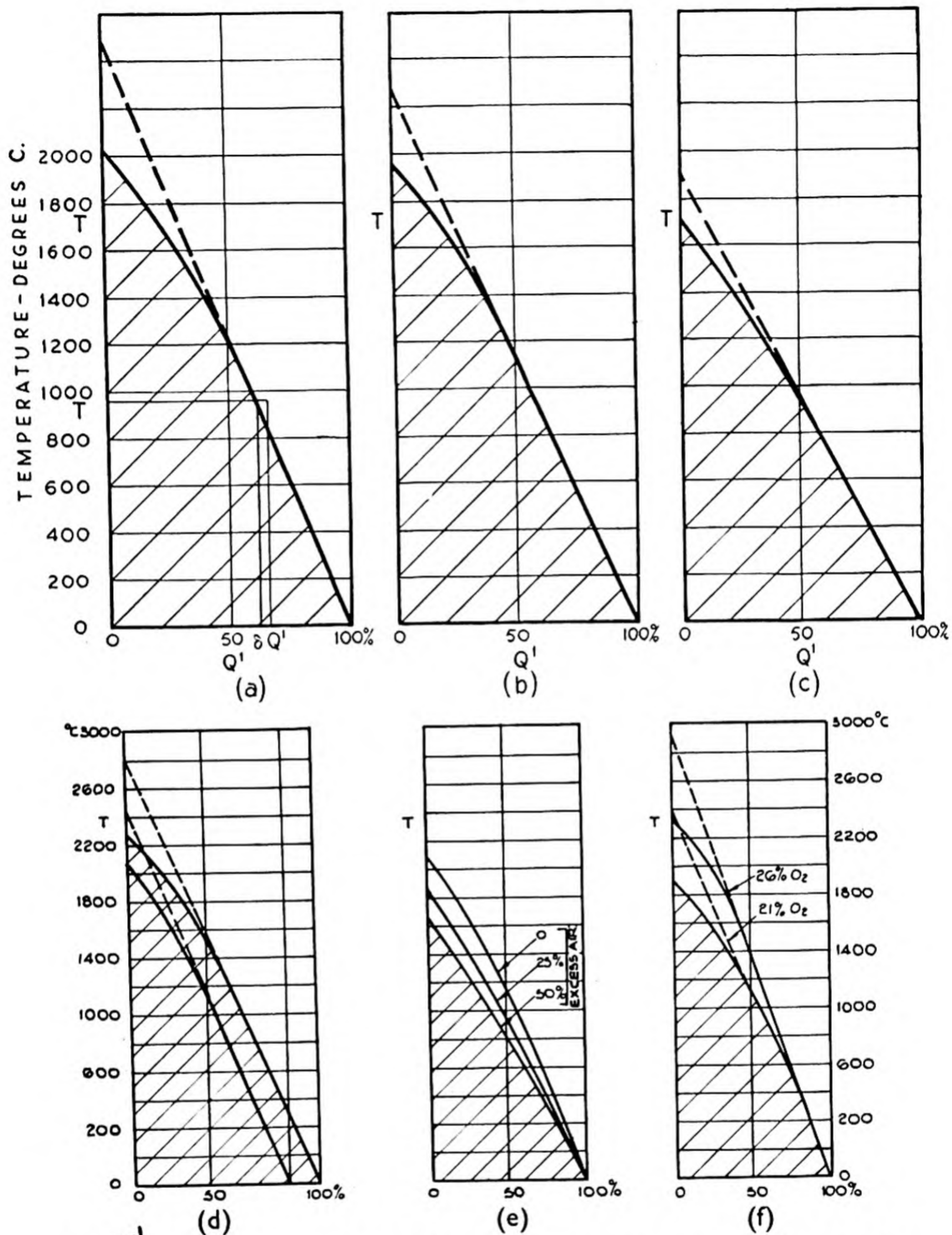


Fig. 2.1. Temperature/Heat Quantity diagrams for the Gases obtained by the Adiabatic Combustion.

(a) Coal with the theoretical air requirements. (b) Town's Gas with the theoretical air requirements. (c) Producer Gas with the theoretical air requirements. (d) Coal with the theoretical air requirements preheated to 500° C. (e) Coal with various amounts of excess air. (f) Coal with oxygen enriched air.



## 2.2.2 THE SCIENCE OF FLAMES AND FURNACES

**2.2.2. The Virtue of Energy.** The type of diagram discussed in the previous subsection does not exhaust the usefulness of the second law of thermodynamics. Such diagrams say what percentage of the total energy is available above any given temperature, but they do not say by how much the quality of this energy will be degraded if it is all transferred to a body at this temperature. The second law of thermodynamics gives also a numerical scale of energy quality which enables this degradation to be assessed. It can be shown that a heat engine working in the most efficient possible way in surroundings at temperature  $T_O$  when fed with a quantity  $Q$  of heat energy all of which is at a temperature  $T_A$ , can only convert an amount  $Q(1 - T_O/T_A)$  into work, the remainder  $(Q \times T_O/T_A)$  being necessarily rejected as heat to the surroundings. This fact provides the definition of energy quality. *The virtue  $V$  of a quantity  $Q$  of energy is defined as the amount of that energy which could be converted into work in an ideally perfect system.* Thermodynamics shows that electrical energy, mechanical work or any form of potential energy are all perfectly convertible, so that this definition is quite precise and definite, giving the relation  $V = Q(1 - T_O/T_A)$ . From this relation it can be seen that the equivalent temperature  $T_A$  of any form of mechanical work is given by  $T_A = \infty$  and also that the virtue of heat at the temperature of the surroundings is zero, a fact which shows at once that the definition is a useful one, since such heat cannot be used for any practical purpose.

The factor  $(1 - T_O/T_A)$  by which  $Q$  is multiplied to obtain the equivalent virtue may be called the quality factor of  $Q$ . The degradation of the energy in being transferred from a body at temperature  $T_A$  to one at a lower temperature  $T_B$  can now be evaluated. Even if such a process is carried out with perfect *thermal* efficiency so that the whole of  $Q$  goes from one body to the other, the loss of virtue is given by  $Q(T_O/T_B - T_O/T_A)$ . Put in other words, this relation states that whereas *before* the heat transfer the heat energy  $Q$  could have been converted ideally into an amount of work  $Q(1 - T_O/T_A)$ , *after* the process the heat  $Q$  could only be converted into an amount of work  $Q(1 - T_O/T_B)$ .

**2.2.3. The Virtue Diagram.** So far only the degradation of energy when all this energy is at one temperature has been considered. In practical cases the energy of a hot gas or hot body is spread out over a whole range of temperatures, as indicated by Fig. (2)6. In this case the virtue of each element of heat energy  $\delta Q$



is different according to the temperature of this element. A simple method of evaluating the virtue of energy which is spread out over a range of temperatures like this is provided by the virtue diagram. The virtue diagram is a modification of the heat availability diagram of Fig. (2)6 in which the ordinate is  $-1/T$  instead of  $T$  ( $T$  must be expressed on the absolute scale whether in Centigrade or Fahrenheit). The negative sign is introduced to give a scale the conventional way up, that is, with the lower temperatures at the bottom. Infinite temperature (mechanical work) appears at a definite position (the zero of the  $1/T$  scale) at the top; this fact is convenient, as it means that this type of energy can be represented on the same diagram. The ordinate scale can be marked off in Centigrade temperatures, although these will not of course appear as uniform steps but will be crowded together at the top. If the points corresponding to the straight line of the perfect gas on Fig. (2)6a are plotted with these axes, a rectangular hyperbola is obtained. The curves on this type of diagram which correspond to those of Fig. (2)6 are shown in Fig. (2)6a-f.

The purpose of this type of diagram becomes clear when it is realised that the area contained between the two ordinates  $Q'$  and  $Q' + \delta Q'$ , see Fig. (2)7a, the curve for the gas and the abscissa  $y_0 = 1/T_0$  (the room temperature line) is equal to  $(y_0 - y)\delta Q'$ ,\* that is, to  $(1/T_0 - 1/T)\delta Q'$ , which is equal to  $\delta V/T_0$  where  $\delta V$  is the virtue corresponding to  $\delta Q'$ . Since  $T_B$  is a constant in any heating process (on the diagram  $1/T_B = \text{distance from bottom abscissa to top one}$ ) the area under the curve between any two values of  $Q'$  ( $Q'_1$  and  $Q'_2$  say) is thus the virtue of the energy  $Q'_1 - Q'_2$ .

The diagram can thus be used to give an easy way of evaluating virtue by evaluation of areas, e.g. by counting squares in any case where the heat content can be calculated as a function of temperature. For example, Fig. (2)7e shows the loss of virtue or degradation of energy that results from diluting combustion gases with air. The method of using this diagram and the results obtainable from it can be illustrated by three of the most important commercial energy-conversion processes. By plotting curves for the material being heated and for the heating gases and counting the squares between them, or by counting the area under each one, the virtue losses in the heat transfer process both due to loss of heat and to degradation of energy can be directly assessed. Similarly, the virtue loss due to dilution of the hot gases by cold gas

\* It is  $y_0 - y$  because the  $y$  scale was made to read downwards.



Fig. 3.1.2. ---  
Virtue Diagrams

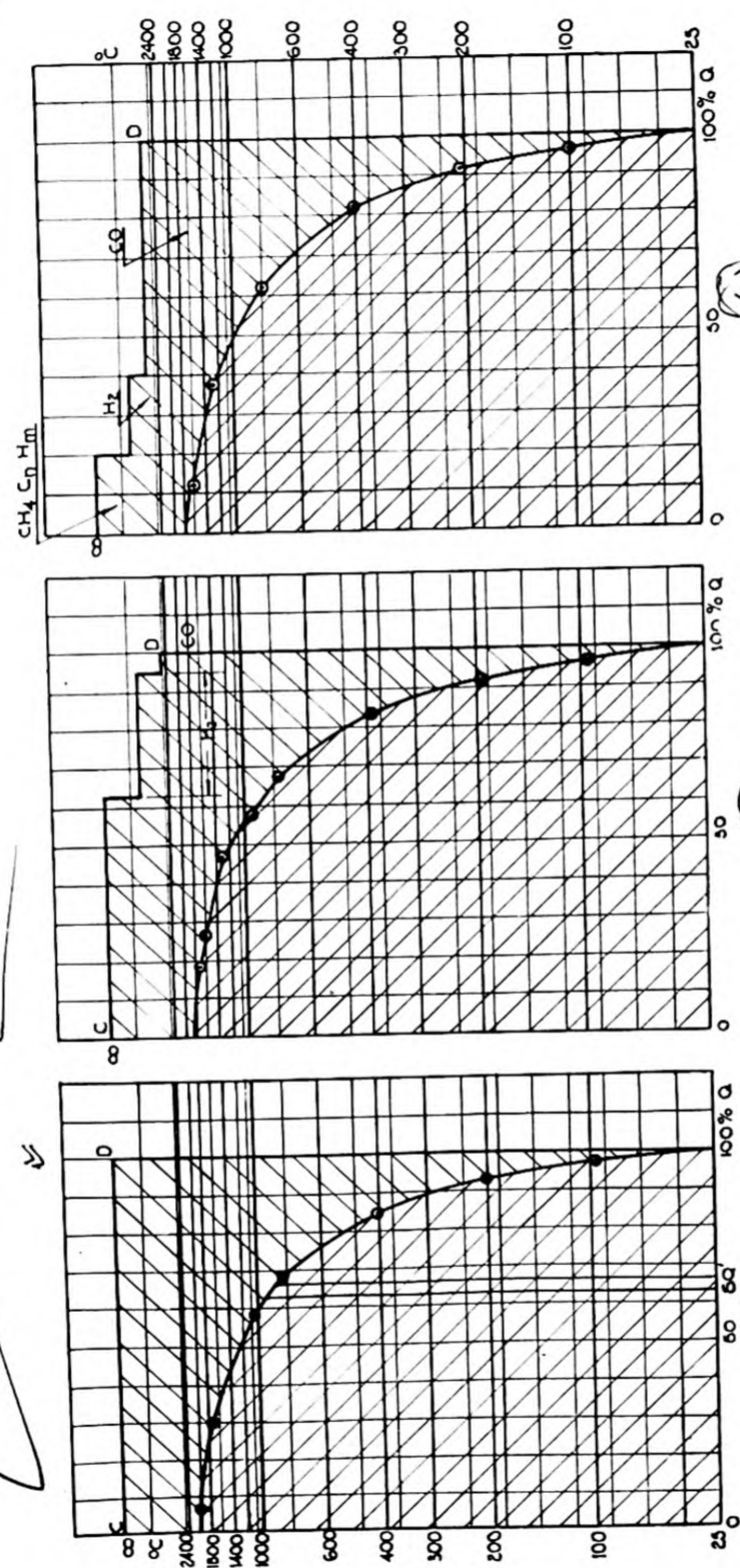
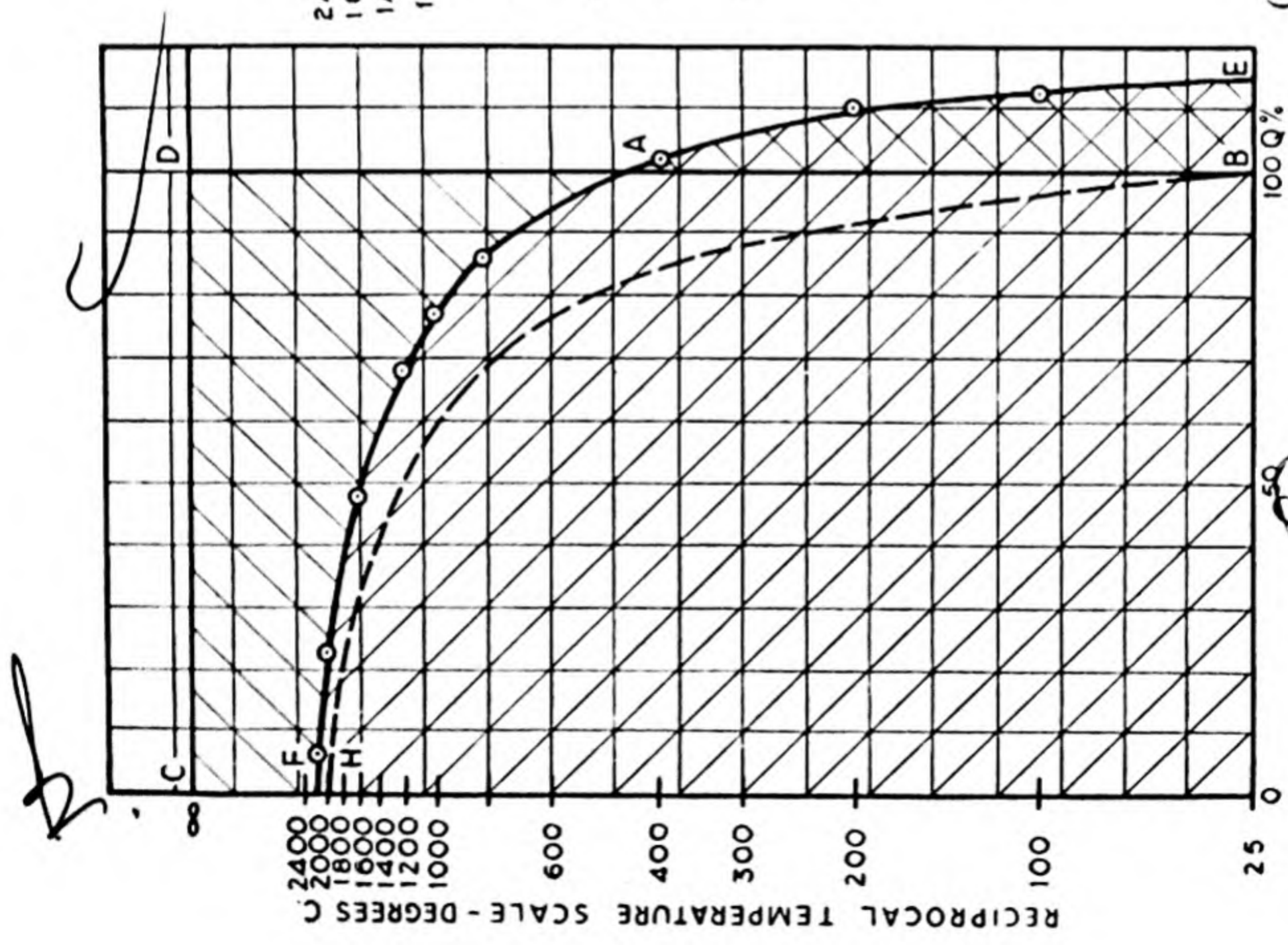
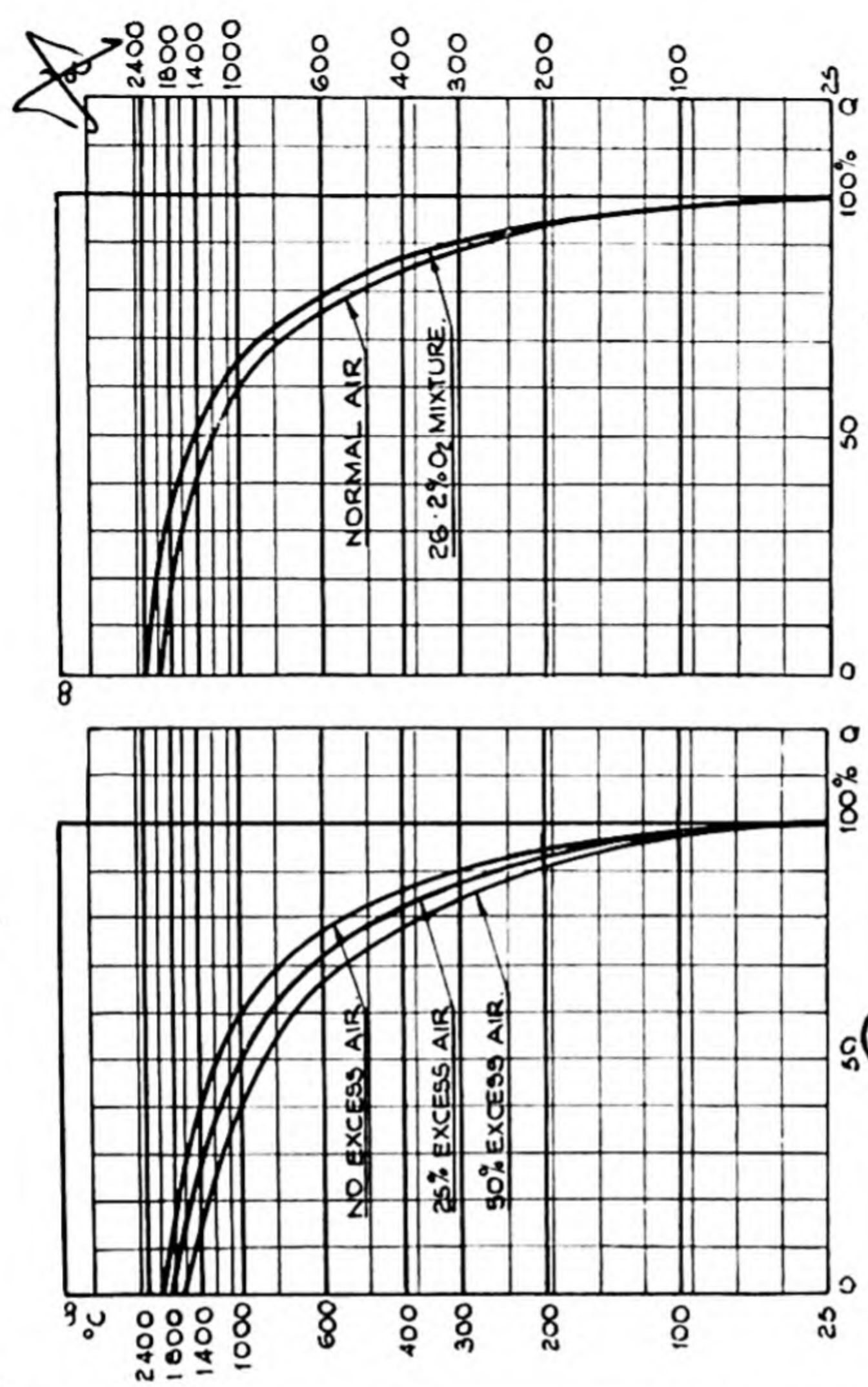


Fig. 3.1.2. ---  
Virtue Diagrams for adiabatic combustion of various fuels. (Headly)  
(Ordinates—temperature °C. (Reversed Reciprocal Scale). Abscissae—percentage of energy of fuel.  
(a) Coal with theoretical air. (b) Town's Gas with theoretical air. (c) Producer Gas with theoretical air.





(d)



(e)

Fig. (e). Virtue Diagrams for adiabatic combustion of various fuels.

Ordinates—temperature °C. (Reversed Reciprocal Scale).  
Abscissæ—percentage of energy of fuel.

- (d) Coal with air preheated to 800° C.
- (e) Coal with various amounts of excess air.
- (f) Coal with oxygen enriched air.

(f)

Fig(3)e



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and consequent lowering of their temperature can be assessed by plotting the curve for the cooler diluted gas and again counting squares between the two curves.

**2.2.4. The Loss of Virtue in a Simple Heat-Transfer Process.** It is worth while considering the loss of virtue in heat-transfer processes in rather more detail, since the degradation of energy in such

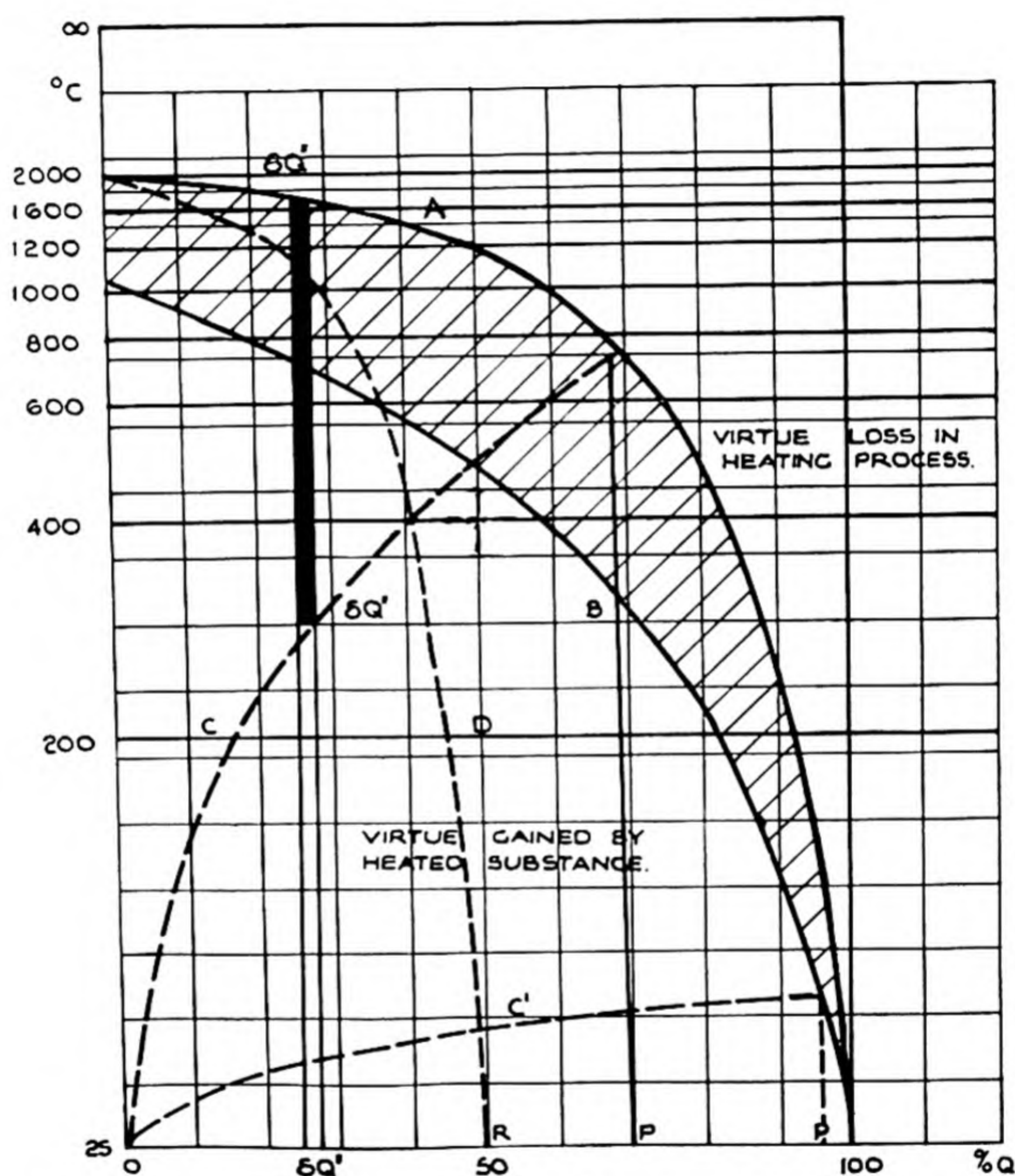


Fig. (2)8. Virtue Diagram for General Heating Process, no Heat Loss.

processes is not sufficiently acknowledged. This represents a straightforward working out of the ideas of Sections 2.2.2 and 2.2.3.

Suppose there is a certain quantity of hot gas,  $m$  units of wt./hr. at a certain temperature  $T_M$ . If its effective specific heat (i.e. specific heat including dissociation energy) is known at all temperatures between  $T_B$  and  $T_M$ , the curve on the virtue diagram corresponding to its energy content can be drawn. Curve  $A$  in

Fig. (2)8 represents such a curve for the hot gas, in which the total heat above room temperature of this gas is called 100%, as in an ordinary heat balance. Suppose, further, that with this gas it is desired to heat a second substance which may be another gas, a solid or a liquid. The heating process can be represented as an ideal heater consisting of a thin wall  $AB$ , Fig. (2)9, dividing two channels, along one of which the heating gas and along the other the heated substance pass. This implies assuming for simplicity that the other walls allow no heat to leak out and the heat transfer area is very large.

The problem is to find by means of the virtue diagram what is the virtue (or second law) efficiency of such a process in comparison with the normal (or first law) thermal efficiency, remembering that

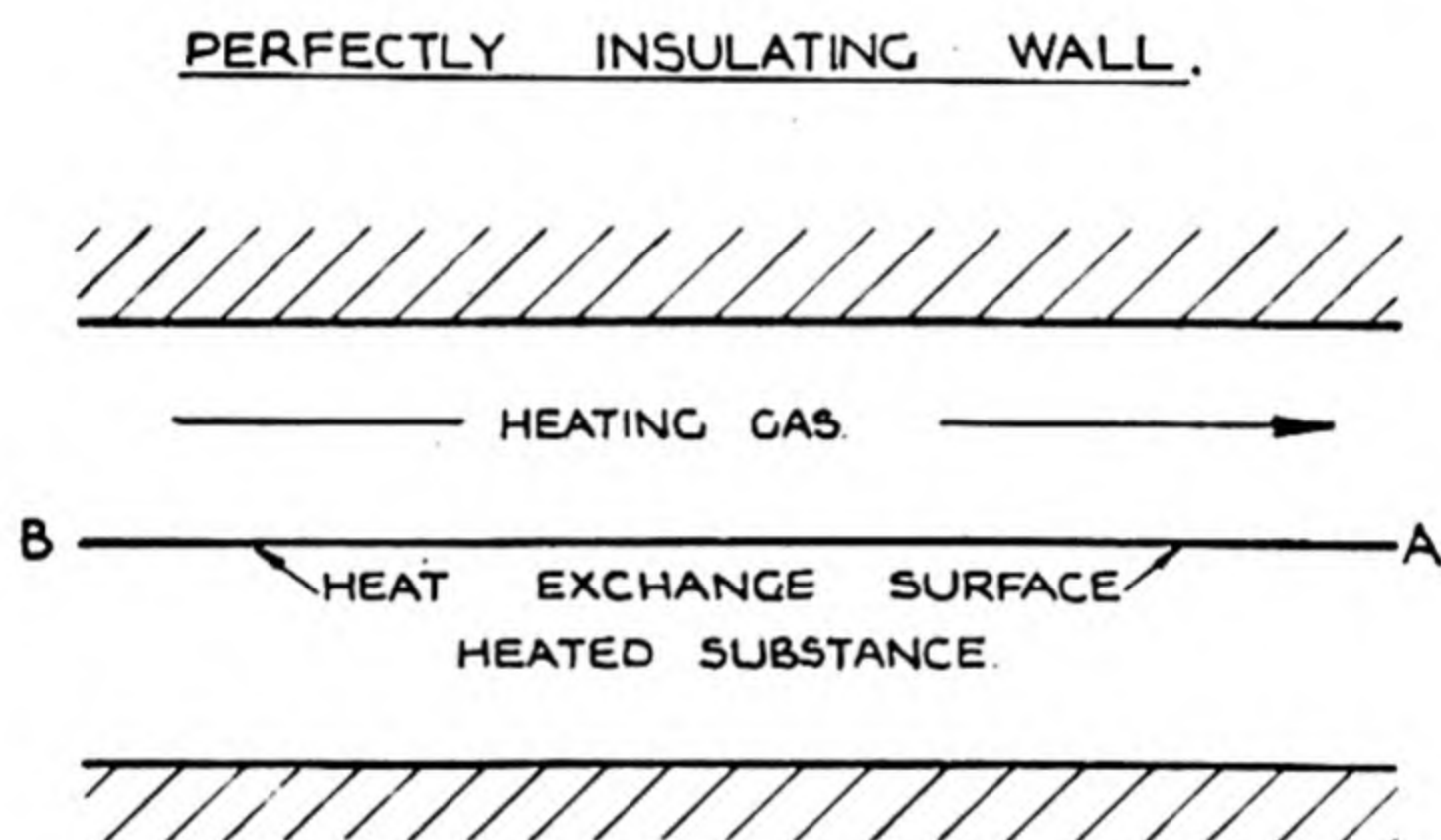


Fig. (2)9. Ideal Heat Exchanger.

if the virtue efficiency is very low *far more energy is being used than would be necessary in a reversible process* of an indirect type, even though the thermal efficiency be 100%. Three cases are of importance :

- (i) Parallel-flow heating ;
- (ii) counterflow heating when the heat capacity of the heated substance is less than that of the heating gas ;
- (iii) counterflow heating when the heat capacity of the former is greater than that of the latter.

(i) *Parallel-flow heating.* In this case when the heated substance, which will be called air, enters at the end  $B$  it cannot possibly leave at  $A$  at a temperature lower than that at which the gas leaves. Hence, only a certain fraction  $OP$ , Fig. (2)8, of the energy of the



## 2.2.4 THE SCIENCE OF FLAMES AND FURNACES

gas can be transferred, where  $P$  is the point on the abscissa corresponding to the intersection of curve  $C$  with curve  $A$ . Curve  $C$  is constructed by plotting  $Q''$  against  $1/T$  where  $Q''$  is the heat energy of the air above room temperature at  $T$ : it is a mirror image of the curve that would be plotted if heat were being taken out of the air because it is so arranged that each element  $\delta Q''$  comes under the element of energy  $\delta Q'$  that flowed through the wall to supply  $\delta Q''$ .

The virtue in the final air is represented by the area between curve  $C$ , the ordinate through  $P$  and the  $1/T_B$  abscissa: that in the final gas is represented by the area between curve  $A$  to the right of the ordinate through  $P$ , this ordinate and the  $1/T_B$  abscissa. This method of heating is thus inefficient on the basis of the first law in that the gas carries quite a lot of heat energy out of the system (unless the amount of air is very much larger than that of the gas, in which case we get a curve like  $C$ ) and also still less efficient on the second law as it degrades very badly the energy that is transferred—all the virtue between  $A$  and  $C$  on the left of  $P$  is lost in the transfer process. This is because the first heat to be transferred undergoes a big temperature drop.

(ii) *Counterflow heating when the heat capacity of the heated substance is less than that of the heating gas.* This case is represented by curve  $D$ . The air cannot absorb all the heat of the gas owing to the low heat capacity of the former (e.g. 5000 ft.<sup>3</sup>/min. of air is being heated by heat exchange from 10,000 ft.<sup>3</sup>/min. of gas of the same specific heat). Hence at the end  $A$  of the heater the gas leaves at a high temperature and the energy which is transferred at this end is badly degraded. This corresponds to the energy flowing from curve  $A$  to curve  $D$  represented by a small area between curves  $A$  and  $D$  on the left of the ordinate through  $R$ . As the air passes along, its temperature comes very close to that of the entering gas, as shown by the convergence of the curves near the ordinate through  $O$ . In this case the virtue of the heat that is transferred is rather better used than in case (i), as the area between the curves is less.

(iii) *Counterflow heating when the heat capacity of the heated substance is greater than that of the heating gas.* This case, which might correspond to 10,000 ft.<sup>3</sup>/min. of air and 5000 ft.<sup>3</sup>/min. of heating gas of the same specific heat, is illustrated by curve  $B$ . All the heat of the gas could be used, but the air cannot be heated above a temperature slightly over half that of the gas (this temperature



can be obtained from the diagram by plotting curve  $B$  on the same energy scale as curve  $A$  and finding the intercept with the ordinate through  $O$ ). Here the virtue utilisation is fairly good because the curves are close together at the lower temperature end, and the thermal efficiency is, with the given assumptions, perfect.

It will be seen from these cases that a direct heating system can only be thermodynamically perfect if, in addition to having no heat losses and infinite heat transfer coefficients, it is counterflow and the heating and heated substances have identical heat-capacity curves. In this case each unit of heat flows across an infinitesimal temperature difference. In the industrial examples it will be shown how by using compound systems with increasing heat-transfer coefficients and separately controlled quantities of heat absorber in the different stages towards the cold end, a rough approximation to the ideal is obtainable in certain cases even when the thermal capacity of the primary heated material is quite different from that of the heating one.

It will also be noticed that the method of representation enables the figures of the thermal balance (first law of thermodynamics) to be read off along the abscissa and the temperatures concerned read off along the ordinate, when the heat capacities and the heat losses are known. In a heating process also the horizontal scale can be directly related to the position in the system at which that particular heat was transferred. These purposes are served as well by the type of diagram of Fig. (2)6, but the latter does not represent the extent of degradation of the energy.

**2.2.5. The Loss of Virtue on Combustion.** The combustion of a fuel is a highly irreversible process, that is to say, it takes place under conditions very far from those of equilibrium. This may be seen from Fig. (2)7d, where a comparison is made between the virtue of the adiabatic combustion gases from coal burnt with and without preheat. The virtue introduced into the system by the preheat of the gases before combustion is only the area  $A, B, E$  and the gases after combustion with preheat contain virtue represented by the area  $F, A, E, B, H$  over and above that contained by the gases from combustion without preheat. Hence the combustion gases from preheated air have extra virtue in excess of that introduced by the preheat (viz. the area  $F, A, B, H$ ). Since virtue can only decrease in any process, it follows that the original coal must certainly have contained more virtue than that represented by the



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upper curve  $F$ ,  $A$ ,  $B$  and hence much more than that represented by the lower curve  $H$ ,  $B$ .

Clearly, then, the virtue of the chemical energy of the fuel is greater than that under any such curve as  $F$ ,  $A$ . It will be a considerable extension of the usefulness of the virtue diagram if it can be used to evaluate this virtue.

The proof of the method by which this can be done<sup>2.5</sup> is not conveniently expressed in terms of virtue and is accordingly omitted from the present monograph. Here, only the result is stated. This is, that the virtue loss on combustion without heat loss, i.e. on converting the chemical energy of the fuel into sensible energy of the gases, is the area between a certain line  $CD$ , Figs. (2)7 a, b, c, d, parallel to and near the line  $T = \infty$  and the curve for the heat capacity of the products of combustion.

The effect of any heat loss which actually takes place during combustion can be represented by regarding it as a separate subsequent reduction of the energy and thus by clipping off the corresponding part of the curve: the virtue loss due to this process is given by the area under the part of the curve so clipped off.

The virtue of the original fuel is thus given by the area bounded by the portion of the  $T = T_B$  abscissa representing the chemical energy of this fuel, by the two ordinates through the ends of this portion of the abscissa, and by the line  $CD$ . In the case of a fuel composed of a mixture of combustible materials the lines for the parts of the total energy corresponding to the different materials can be drawn at the appropriate heights. Examples of this construction are shown in Figs. (2)7 a, b, c.

Among materials burnt with air, the correction is of importance only in the case of  $\text{CO}$  and  $\text{H}_2$ , since these are the only fuels for which there is a significantly great change of volume on combustion. The  $\text{CO}$  line lies below  $T = \infty$  by an amount corresponding to  $1/T_A = 3.11 \times 10^{-4}$ , while that for  $\text{H}_2$  lies below  $T = \infty$  by  $1/T_A = 1.84 \times 10^{-4}$ .

The significance of this correction may be illustrated by the reduction of  $\text{CO}_2$  in hot gases passing through a fuel bed. This reaction absorbs thermal energy to produce an equal amount of chemical energy, which at first sight seems to be an increase in virtue and so to contradict the second law.

Actually, however, the difficulty is cleared up owing to the fact

<sup>2.5</sup> THRING, M. W.: "The Virtue of Energy, its Meaning and Practical Significance," *J. Inst. Fuel.*, 17, p. 116 (1943-44).



that CO has less virtue than the carbon consumed in the reaction. This is illustrated by the case of combustion gas of composition 21%  $\text{CO}_2$ , 89%  $\text{N}_2$  at a temperature  $1800^\circ \text{C}$ . coming into contact with carbon and reacting to produce gas of composition 12%  $\text{CO}_2$ , 12% CO, 76%  $\text{N}_2$  at a temperature (point R) of about  $1350^\circ \text{C}$ . Fig. (2)10 deals with this case.

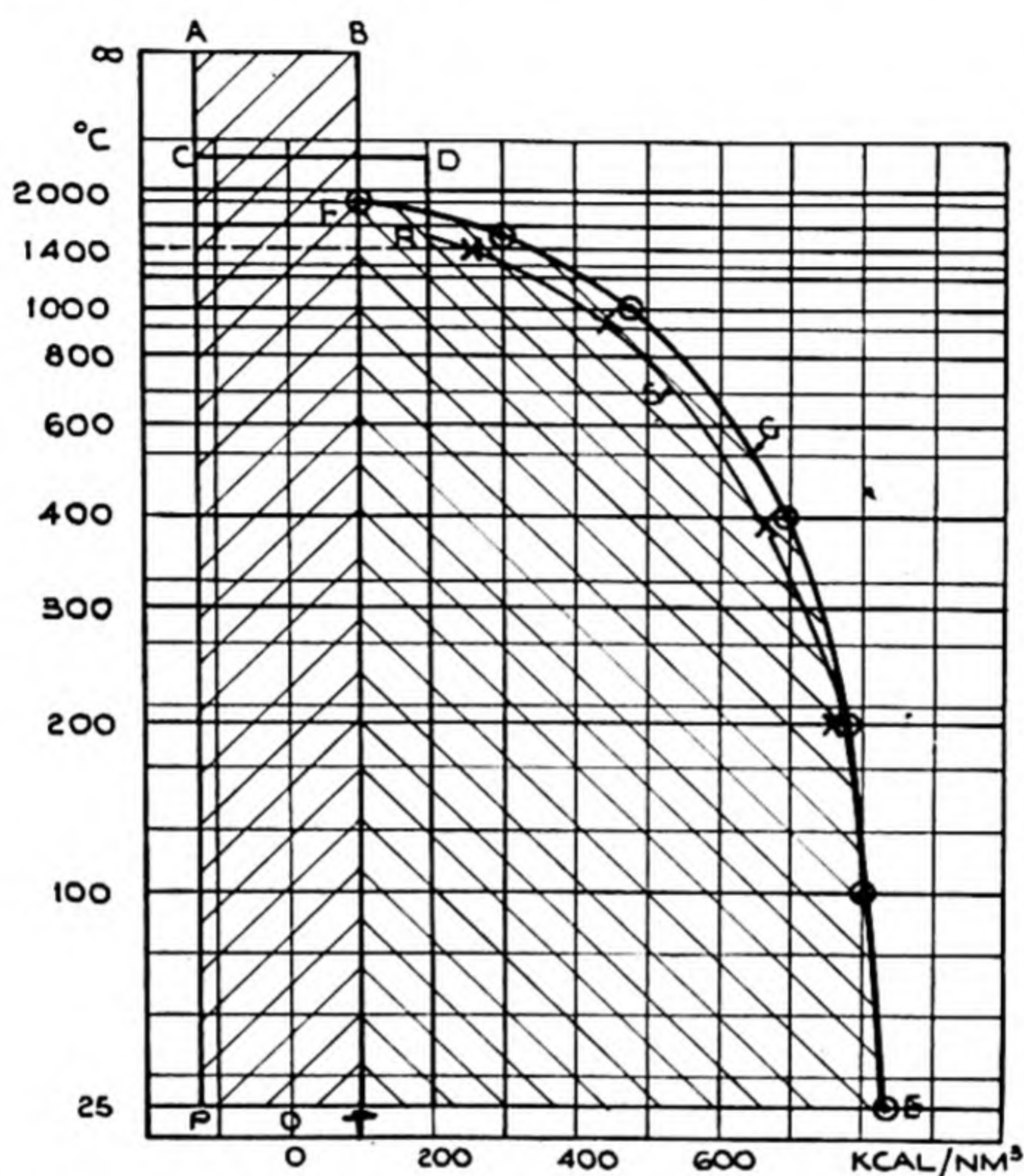


Fig. (2)10. Virtue Diagram for  $\text{CO}_2$  Reduction Process.

The units of  $Q'$  in this case are  $\text{k.cal./nm}^3$  of the original gas. The area PABQ represents the virtue in the unburnt carbon and OFGE that in the hot gas before the reaction,\* while PCDT represents the virtue of the CO, and TRSE that of the hot gas after the reaction. Expressed in squares: ✓

Virtue before reaction, in carbon	.	.	.	35.5
" " " in sensible heat	.	.	.	81
Virtue after reaction, in CO	.	.	.	46.4
" " " in sensible heat	.	.	.	66

If CO had the same virtue as C (for the same heat quantity) the

\* The curves have been calculated by means of the Rosin-Fehling IT diagram.



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third figure would be 51.2 and the loss of 15 units from sensible heat would not be sufficient to make up the gain in chemical virtue.

Fig. (2)11 shows a second example—a dry gas producer operating on coke. The units for  $Q$  in this case are 1 cm.=1000 k.cal./kg. of the fuel, and it has been assumed that the producer makes a gas of composition 30·2% CO, 2·8% CO<sub>2</sub> and 67% N<sub>2</sub>, and that it loses 5% of the energy of the coke by heat losses. The final

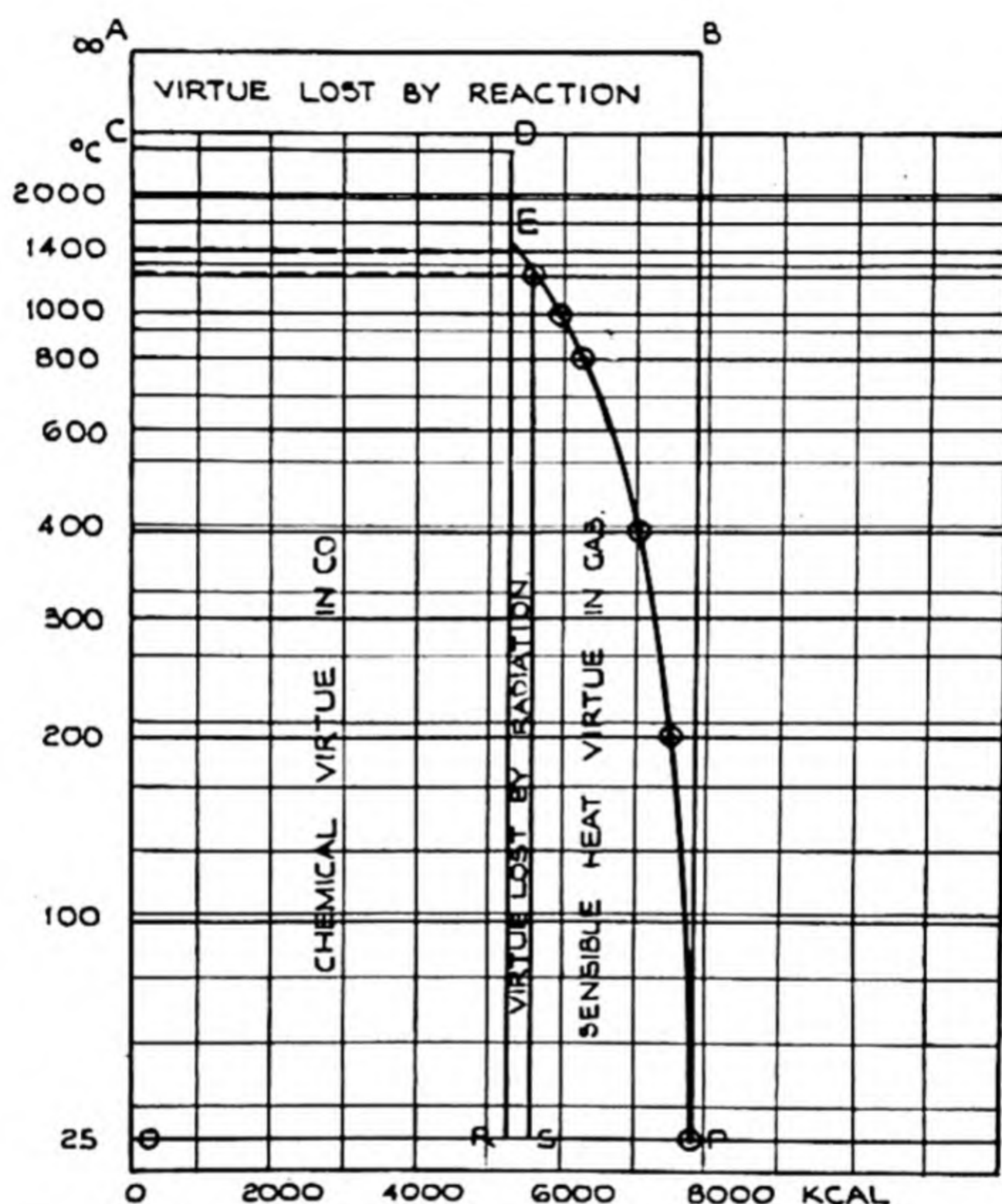


Fig. (2)11. Virtue Diagram for Dry Gas Producer.

temperature of the gas (point *E*) is just over 1200° C. Counting squares gives the following result :

	Squares	% on original fuel
Virtue in original fuel <i>ABOP</i>	132	100
Chemical virtue in CO	81	62
Virtue in sensible heat of gas	22	29
Virtue lost by radiation	5	4

Hence the hot gas virtue efficiency is 91 % compared with a hot gas first law efficiency of 95% ; the corresponding cold gas figures are 66% and 62%.

The virtue losses on combustion under the various conditions of Fig. (2)7 can now be assessed. The results are given in Table 2.3. The only significant difference in the virtue loss on adiabatic combustion in this table is that which occurs when there is appreciable excess air. Although change of fuel and the use of oxygen-enriched air alter greatly the amount of energy available at any given high temperature (enriching the combustion air with  $O_2$  doubles the amount available at  $1850^\circ C.$ ), they do not alter the virtue change greatly because of the  $1/T$  factor which reduces the relative importance of energy variations at high temperatures.

TABLE 2.3. VIRTUE LOSSES ON COMBUSTION

	Coal	Town's Gas	Producer Gas	Coal with 50% $O_2$ excess air	Coal with $O_2$ enriched air
Virtue in original fuel (squares)	168	164	156	<u>168</u>	<u>168</u>
Virtue in hot gases before heat loss (squares)	122	120	118	112	124
Virtue loss on adiabatic combustion as percentage on virtue of original fuel	27%	27%	24.5%	<u>33%</u>	<u>26%</u>

### 2.2.6. Examples of Virtue Diagrams for Industrial Furnaces.

(a) *Cement kiln.\** Fig. (2)12 shows the virtue diagram constructed for such a kiln from the following data :

- Fuel . Coal of 12,600 B.Th.U./lb. (7000 k.cal./kg. or C.H.U./lb.).
- Excess air . 2.5%.
- Preheat . 85% of air for combustion heated to  $400^\circ F.$  ( $234^\circ C.$ ).

From these figures the uppermost curve is calculated for the adiabatic combustion of the fuel.

\* This consists of a long, gently sloping cylinder, fired at the lower end with pulverised coal, the gases from which leave the upper end together with moisture and  $CO_2$  from the cement. The cement slurry is fed in at the upper end of the kiln and passes out at the lower into a second barrel where it gives up some of its heat to air which is supplied to the main kiln for combustion. This is shown on the inset to Fig. (2)12. It is thus a very good example of a genuine counterflow process.



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Output . 3 lb. of dry clinker/lb. of coal.  
 Feed . 1.56 lb. dry slurry (containing 0.52 lb. of  $\text{CO}_2$ ).  
 1 lb. of  $\text{H}_2\text{O}$  per lb. of dry clinker.

Heat capacity of clinker at :

0–810° C.—604 B.Th.U./lb. (336 C.H.U./lb.).

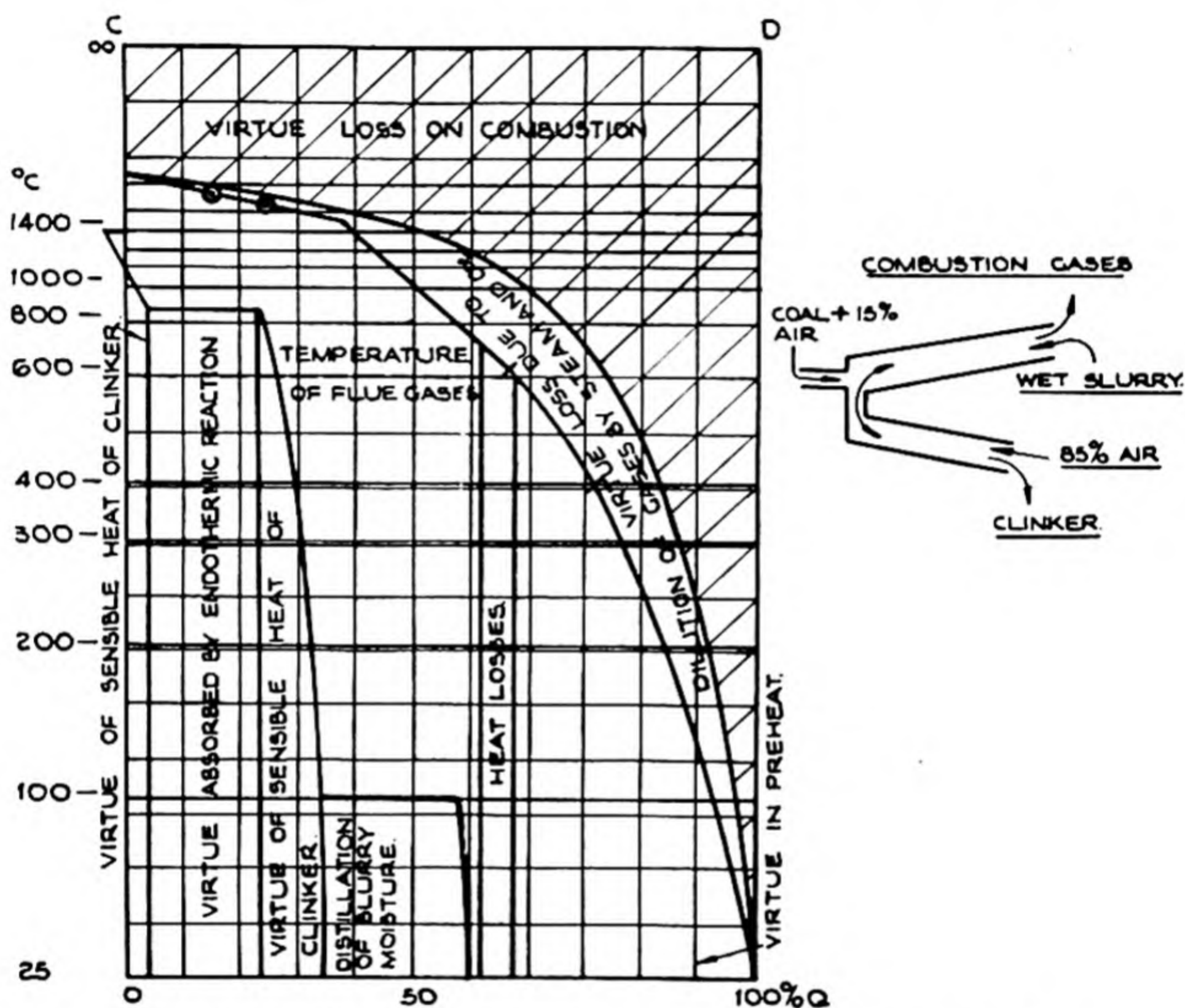


Fig. (2)12. Virtue Diagram for Actual Rotary Cement Kiln.

Heat absorbed by reaction at :

810° C.—812 B.Th.U./lb. (451 C.H.U./lb.).

810° C.—1370° C.—287–180=107 B.Th.U./lb. (594 C.H.U./lb.).

Total : 1523 B.Th.U./lb. (846 C.H.U./lb.).

From these figures the  $1/T-Q$  curve for the clinker can be calculated using the energy of the fuel as the basis for  $Q$ . The virtue enclosed by this curve is shown subdivided into the virtue of distillation of the moisture, that which appears as the sensible heat in the final clinker and that absorbed by the endothermic reaction  $\text{CaCO}_3 \rightarrow \text{CaO} + \text{CO}_2$ . The subdivision of the 25° C. line

corresponds to the conventional heat balance based on the first law of thermodynamics, viz. :

Sensible heat content of clinker	17
Heat absorbed by reaction	19
Heat absorbed in boiling slurry water	26
Heat losses	5
Heat in exit gases	40
	<hr/> 107 <hr/>

The excess 7% of this energy is that recycled from the sensible heat content of the clinker by the air preheat. The diagram is of particular interest, however, in the way it shows the effect of the dilution of the flue gases by the  $\text{CO}_2$  and  $\text{H}_2\text{O}$  from the wet slurry. The  $Q-1/T$  curves are calculated as before, allowing for the extra heat capacity per lb. of fuel, and this shows the virtue loss due to cooling the flue gases, and enables the temperature of the gases leaving the kiln to be read directly from the revised curve.

The actual virtue balance for Fig. (2)12 is as follows :

	Actual Kiln (Squares)	
Virtue in original fuel	168	
Virtue loss on combustion	39	
Total virtue in hot burnt gases	131	
Virtue loss due to dilution	12.6	
Virtue recycled <i>from products</i> by preheat	2.3	
Total virtue supplied to clinker	47	{ Reaction, 23 Sensible, 15 Moisture, 9
Virtue in heat losses	5.6	
Virtue in exit gases	29	
Virtue loss in heat transfer	38	
Virtue efficiency	28%	

From this table it can be concluded that the virtue loss in heat transfer in a cement kiln largely arises from the extent to which it is used as a boiler compared to the amount of high temperature heating.

(b) *Open-hearth furnace.* An average typical heat balance for a regenerative open-hearth furnace is shown in Fig. (2)13,<sup>2.6</sup> and

<sup>2.6</sup> *Mitt. der Wärmestelle*, No. 53, p. 7 (1927).

KERR, SINCLAIR, J. : "Fuel Economy and Control in the Basic Industries with Special Reference to Iron and Steel Works," *J. Inst. Fuel*, 5, p. 154 (1931-32).

WELLMAN-SEEVER-MORGAN Co., *The Open-Hearth*, New York, 1920 (U.P.C. Book Co., Inc.).

COLCLOUGH, T. P. : "Regeneration in Glass Melting Furnaces," *Trans. Soc. Glass Tech.*, 16, p. 57 (1932).



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the corresponding virtue diagram in Fig. (2)14. The producer gas enters the system at a temperature of just under  $400^{\circ}\text{C}$ . with 10% of its energy in the form of sensible heat, and it has been assumed

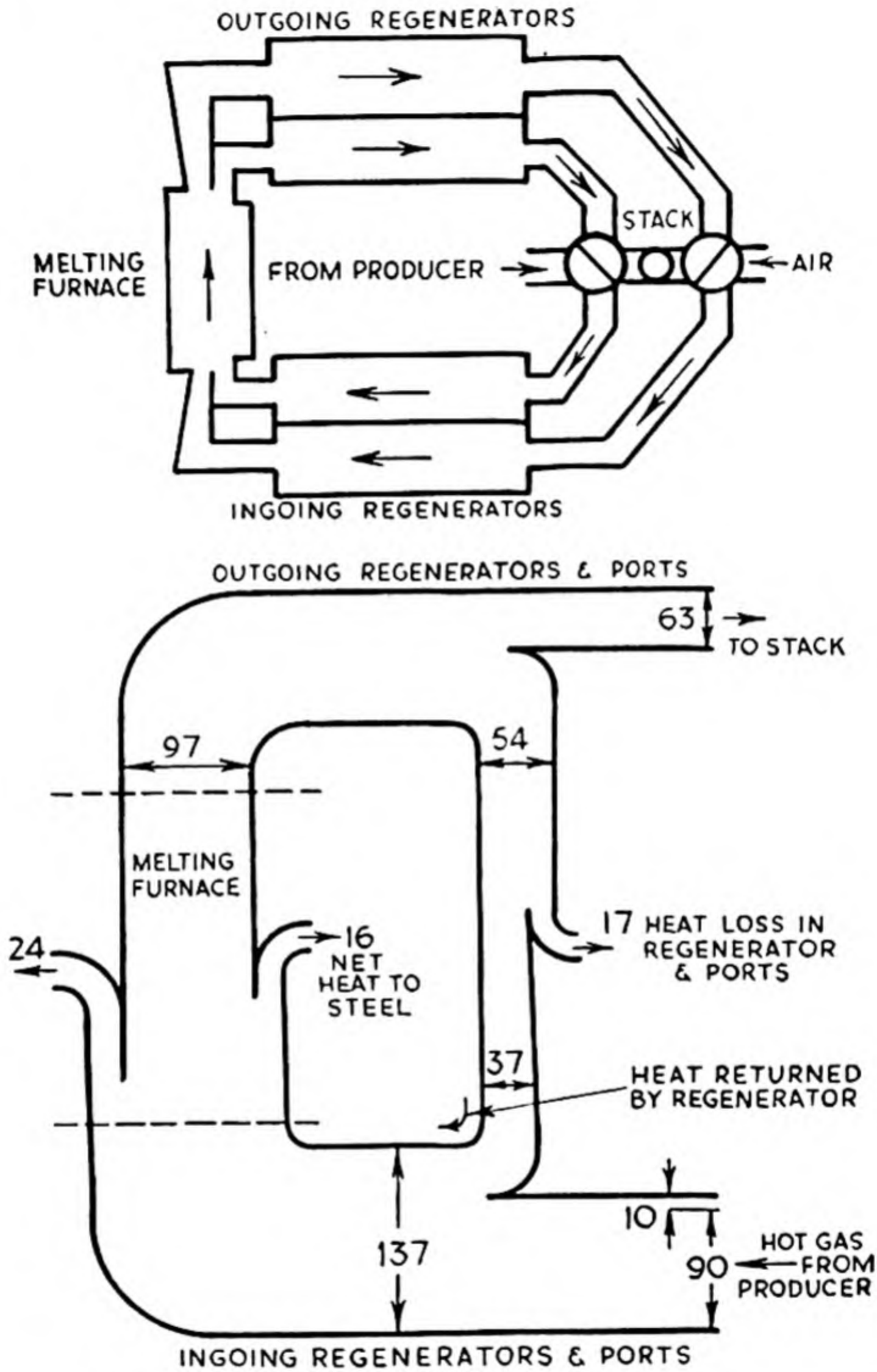


Fig. (2)13. Sankey Diagram of the Heat Balance for an Open-Hearth Furnace.

to have the same composition as the gas of Fig. (2)7c. Its original virtue at the gas valve is thus represented by the area  $CDEAO$  and its total heat by the length  $OA$ .

The adiabatic combustion curve  $URFJI$  is calculated from the total energy of the gas before combustion (137% compared to

90% chemical energy) assuming 25% excess air, while  $AH$  is the corresponding curve for the gases without preheat, i.e. starting from the 100% point, which is used to give the temperature  $H$  of the

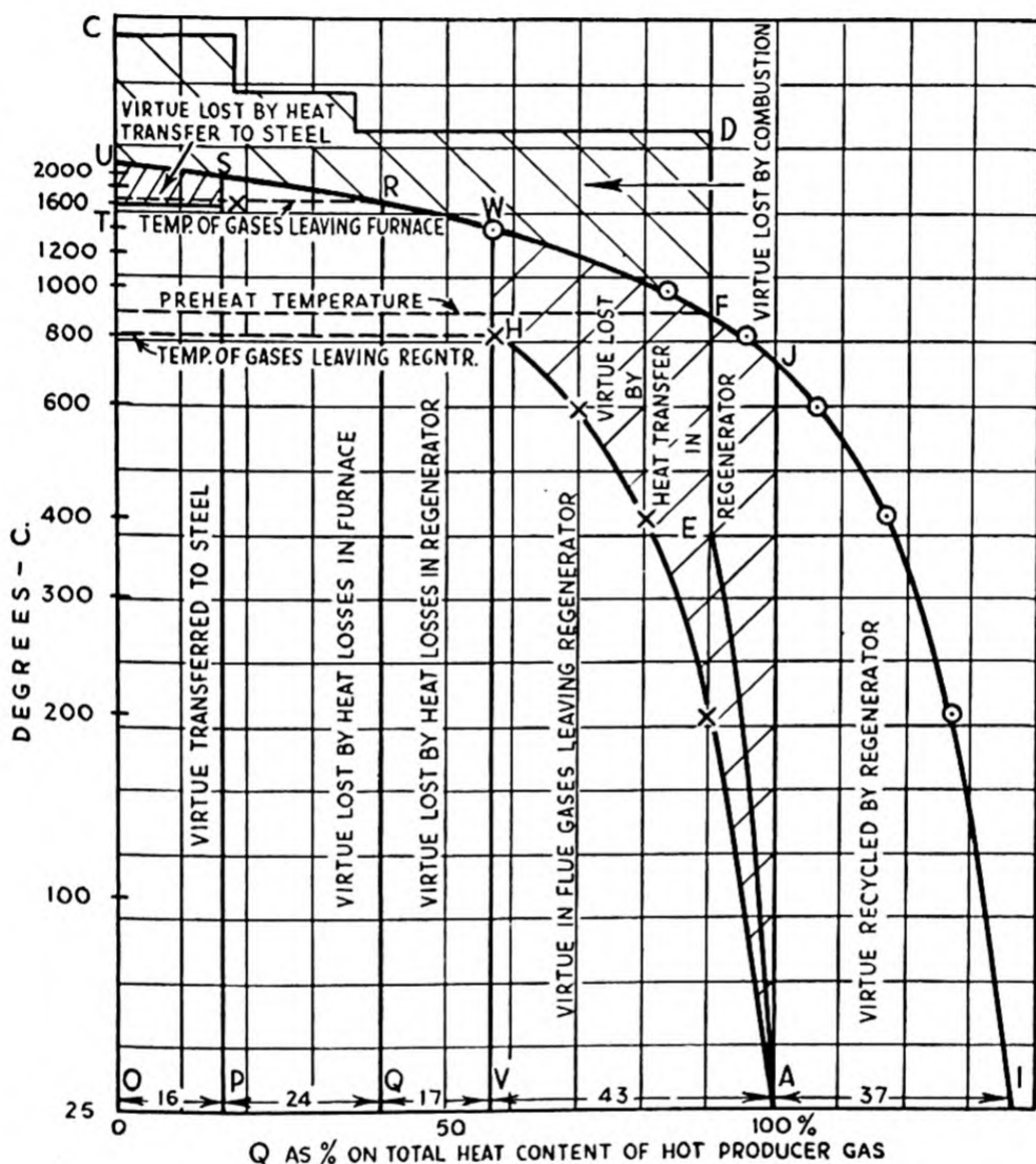


Fig. (2)14. Virtue Diagram for Actual Open-Hearth Steel Melting Furnace.

gases leaving the regenerator and the virtue loss by heat transfer in the regenerator. The point  $F$  corresponds to the mean air-gas preheat temperature, as this includes the original sensible heat of the gas.



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The virtue figures for this case are as follows :

	Fig. (2) 14, Actual Furnace 146	Fig. (2) 15, Idealised Furnace 156
Virtue in original fuel, <i>CDEAO</i>		
Loss of virtue on combustion <i>CDEAO</i> — <i>URFJAO</i>	9	10
Total virtue in gases after combustion in- cluding virtue recycled <i>URWFJIO</i>	167	245
This is split up into :		
1. Virtue recycled <i>JIA</i>	29	99
2. Virtue in exit gases <i>HAV</i>	35	0
3. Virtue loss due to irreversible heat transfer in regeneration <i>WJAH</i>	20	0
4. Heat losses <i>SWVP</i> { in regenerator in furnace	34 34	0 0
5. Virtue loss due to heat transfer in furnace <i>USXT</i>	0.8	5.5
6. Virtue transferred to steel <i>TXOP</i>	23	141

The column for Fig. (2)15 refers to an idealised furnace without

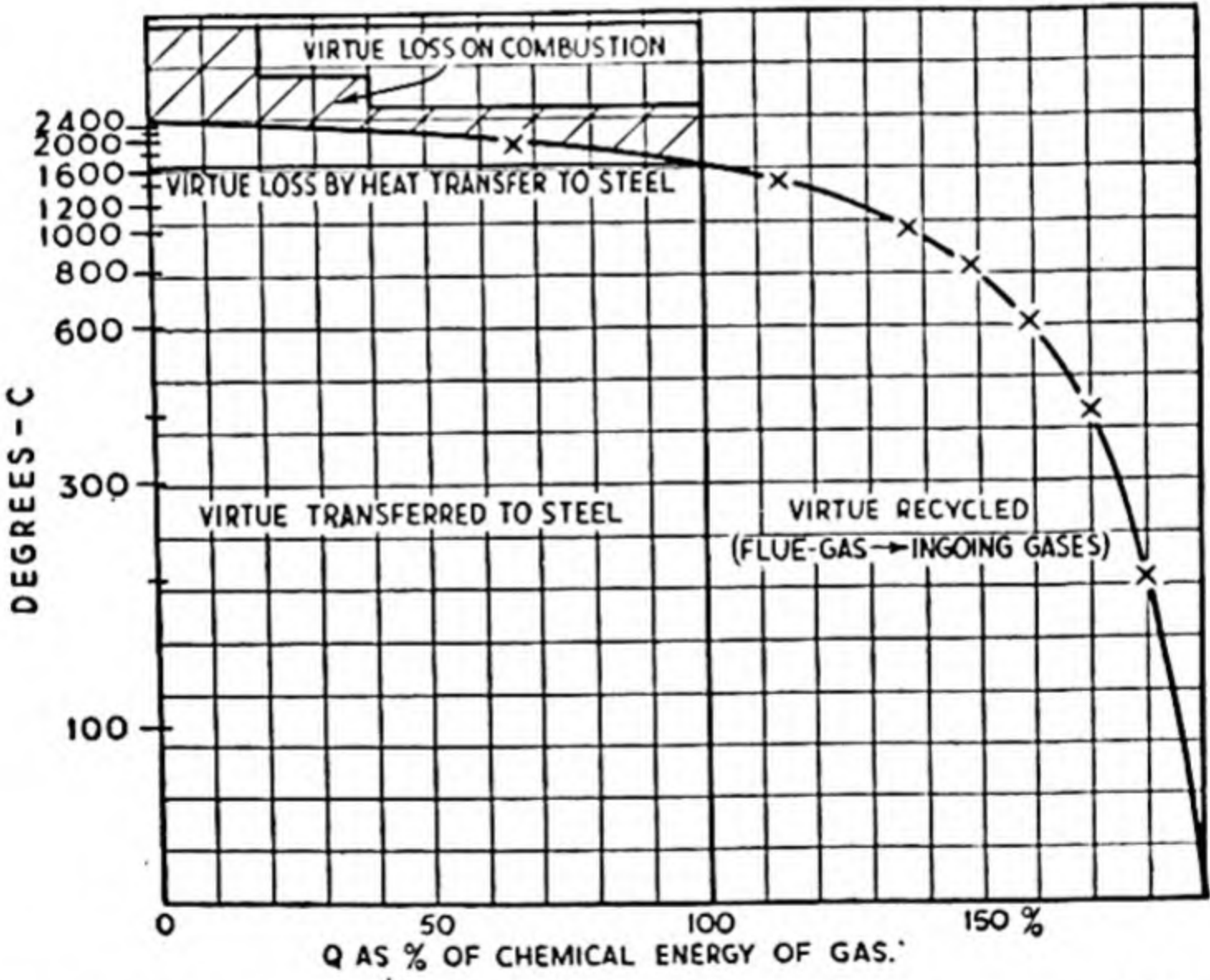


Fig. (2)15. Virtue Diagram for Idealised Open-Hearth Furnace with no excess air, no heat losses, perfect heat transfer in Regenerators and using Cold Gas.

heat loss for excess air and with perfect heat transfer in the re-generators.

This example illustrates the case where the heat is required mainly at a very high temperature so that the virtue loss due to heat transfer is very small, but on the other hand the amount of heat in the gases leaving the heating chamber is high. It is thus the exact opposite of the case of a boiler which has a very good first law



efficiency but a large virtue loss because the virtue of energy in steam is low. The very low thermal efficiency which would result from this state of affairs is, however, overcome by recycling the energy in the gases below the furnace temperature by means of a preheater. In this way the virtue loss on combustion is also very much reduced and the whole system represents good practice so far as the second law of thermodynamics is concerned, the theoretical virtue utilisation being very high. Actual furnaces, as represented by Fig. (2)14, fall short of the ideal because of the difficulty of effective insulation and of having sufficient heat transfer surface.

**2.2.7. Application of Thermodynamics to the Case where the Preheat of the Fuel must not exceed a Certain Temperature.** Larsen and Siddall<sup>2.7</sup> have discussed the effect on the maximum possible efficiency of a regenerative furnace of assuming that certain fuel mixtures have an optimum preheat temperature to give maximum luminosity and that certain constituents of fuel gases (particularly methane) can absorb heat in the incoming regenerators by endothermic reactions. This implies that some fuels can make more use of regeneration than others, so that the former can give a higher limiting efficiency if perfect regenerators are available and can make more use of large regenerators than fuels in which it is necessarily impossible to recycle a considerable fraction of the outgoing energy, however large and efficient the regenerators are.

They calculate the percentage of the heat which can be produced as high-temperature heat, i.e. over  $1570^{\circ}\text{C}$ . if the regenerators are perfectly efficient, i.e. have no heat losses and transfer all the heat allowed by the heat capacities of the ingoing and outgoing gases. They also calculate, for a chosen large checker capacity and a chosen small checker capacity, the actual preheat and waste-gas temperatures which will be obtained; these calculations are made by means of an electrical analogue method for regenerator effectiveness similar to those discussed below in the sections on heat transfer. The key assumptions are that producer gas must not be heated above  $1010^{\circ}\text{C}$ .<sup>\*</sup> to avoid excessive loss of flame luminosity, whereas with a mixture of coke oven gas and blast furnace gas the methane and hydrocarbons can react with the  $\text{CO}_2$  and  $\text{H}_2\text{O}$  endothermically,

<sup>2.7</sup> LARSEN, C. B. D., and SIDALL, S.: "Theoretical Limiting Efficiency of Various Fuels in the Open-Hearth Furnace," *Iron and Steel Engineer*, pp. 76-96 (Dec. 1945).

<sup>\*</sup> This assumption is necessary for the achievement of definite results but it is only justified when the gas producers are operated to give a certain moisture content in the gas as discussed below.



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thus giving a greater heat-absorbing capacity in the regenerators. Hence, provided the amount of coke-oven gas in this mixture is sufficient, there will be some hydrocarbons to spare and therefore luminosity will not be impaired whatever the preheat temperature. They therefore conclude that a mixture containing 12% methane, composed of 35% coke-oven gas and 65% blast-furnace gas, is the ideal fuel, since with it perfect regenerators could return 93% of the heat leaving the furnace. In this case the endothermic reactions in the checkers give 7% increase in the chemical energy of the fuel and a 22% increase in its volume. As one would expect with this fuel mixture, the efficiency drops steadily if the checker capacity is inadequate: thus with 20 ft. deep checkers 70% of the ingoing fuel energy is available as high-temperature heat, while with 12 ft. checkers only 57% is available.

Owing to its large content of nitrogen and its limiting preheat temperature of  $1010^{\circ}\text{C}$ ., straight producer gas can benefit least from larger regenerators and even with perfect regeneration can only give 50% of its heat available to the high-temperature zone. When, however, 10% of coke-oven gas is mixed with it, it can be preheated to a higher temperature without loss of all the luminous constituents and the amount of high-temperature heat can be raised to 62%. Oil comes intermediate between premixed coke-oven and blast-furnace gas on the one hand and producer gas on the other. It is shown that provided the checkers were large enough it would even be possible to make steel with neat blast-furnace gas, the available high temperature heat being slightly lower than that with producer gas. The results for different fuels are summarised in Table 2.4.

These calculations are very interesting since they represent an attempt to apply the two laws of thermodynamics to the open-hearth system and to give from this application a reliable comparison of the potentialities of the different fuels. As in all applications of thermodynamics, however, the results do not give time effects, and in fact these may alter the relative values of the fuels somewhat. Moreover it is not really fair to state an exact limiting temperature for the preheat of producer gas since the loss of flame luminosity resulting from excessive preheat is the result of the acceleration of the water gas reaction  $\text{C} + \text{H}_2\text{O} \rightarrow \text{CO} + \text{H}_2$ . This reaction results in the disappearance of carbon which arose due to tar dissociation and will increase at higher temperatures, but if the producers are operated with a smaller steam addition so as to give



a smaller amount of undecomposed steam in the gas, then clearly the limiting temperature for producer gas preheat can be increased. It is also possible to some extent to make up for the limiting preheat of the producer gas by increasing the air preheat. (The author is indebted to Monsieur G. Husson of the Institut de Recherches de la Sidérurgie for pointing out this limitation).

There is little doubt, however, than an experimental investigation of the amounts of heat going to the checkers and the quantity of waste gas divided between them, the degree by which different fuels fall below their different efficiencies obtained, would give results of

TABLE 2.4\*

## COMPARISON OF CALCULATED EFFICIENCY OF SEVERAL FUELS

	% high-temperature or available heat		Mean temperature of outgoing gases at air inlets	
	Depth of checkers—ft.			
	20	12	20	12
Premixed coke-oven and blast-furnace gas	70	57	770	1140
Premixed natural and blast-furnace gas	66	—	760	—
Oil	64	57	1090	1340
Producer gas plus 10% coke-oven gas, premixed	62	56	1120	1320
Coke-oven gas	62	56	1170	1355
Coke-oven gas and blast-furnace gas, not premixed	59	53	1130	1320
Producer gas	55	50	1360	1525
Blast-furnace gas	53	—	960	—

\* 2.7. Table IX, LARSEN and SIDALL, *Iron and Steel Engineer*, pp. 76–96, Dec. 1945).

great value in open-hearth operation since the actual efficiencies obtained with different fuels could then be compared with the theoretical.

### 2.2.8. The Calculation of Reduction and Oxidation in Furnaces.

A further application of the second law of thermodynamics which is of considerable importance to furnace design and operation relates to its use to calculate the variation of chemical equilibria with temperature. Since it is necessary to discuss this calculation in relation to combustion reactions it is dealt with in the next chapter (3.1.3), and the reduction and oxidation of the furnace charge is also discussed in that section.



**2.2.9. The Use of Oxygen in Iron and Steel Furnaces.** A treatment of the use of oxygen for combustion of the fuel in open-hearth furnaces and blast furnaces and of carbon in molten iron to make steel can be made using the same methods as those of Larsen and Sidall, namely, applying the second law of thermodynamics in the form of the requirement that heat cannot be transferred spontaneously from a colder to a hotter body, so that the combustion gases must be hotter than the charge at any point.

In order to achieve the best possible case the heat transfer to the charge must be taken to be perfect so that the gases leave at  $T_1$ , the furnace must be perfectly insulated, and air and fuel must be in the stoichiometric ratio. It is also useful to regard the combustion gases as perfect, i.e. with a constant specific heat.\* For such an ideal system it is at once evident that the heat equivalent of the fuel which must be burnt to produce the quantity  $H_1$  of heat at a temperature  $T_1$  is given by

$$H_m = \frac{\theta_m}{\theta_m - \theta_1} H_1 \quad \dots \quad (2/1)$$

where  $\theta_m$  is the difference between the theoretical temperature of combustion and room temperature, while

$\theta_1$  is the difference between  $T_1$  and room temperature.

Since the theoretical temperature of combustion of most rich fuels with air, not allowing for dissociation or increase in specific heat above  $1600^\circ \text{C.}$ , is about  $2200^\circ \text{C.}$ , it follows that the theoretical heat equivalent in fuel to produce a quantity  $H_1$  of heat at a temperature  $T_1$  is given roughly by

$$\frac{H_1}{1 - \frac{\theta_1}{2200}} \quad \dots \quad (2/2)$$

Correspondingly, the ideal efficiency of a combustion system to produce heat at a temperature  $T_1$  is given roughly by

$$1 - \frac{\theta_1}{2200} \quad \dots \quad (2/3)$$

\* In practice this assumption is reasonably true, apart from the effects of dissociation which come in at temperatures above  $1600^\circ \text{C.}$  Since, however, the temperature  $T_1$  is rarely above  $1600^\circ \text{C.}$  and dissociation is reversible, the correct evaluation of the fractional heat lost below  $T_1$  is correctly given by using a linear relationship passing through the theoretical maximum temperature  $T_m$  corresponding to no dissociation. This point may be seen with reference to Fig. (2)16, where the available heat in the gases

This same analysis may readily be extended to the case where a preheater absorbs a fraction  $\eta$  of the heat leaving the furnace, returns this as preheat, and so raises the theoretical temperature of combustion. In this case it may be shown that the theoretical

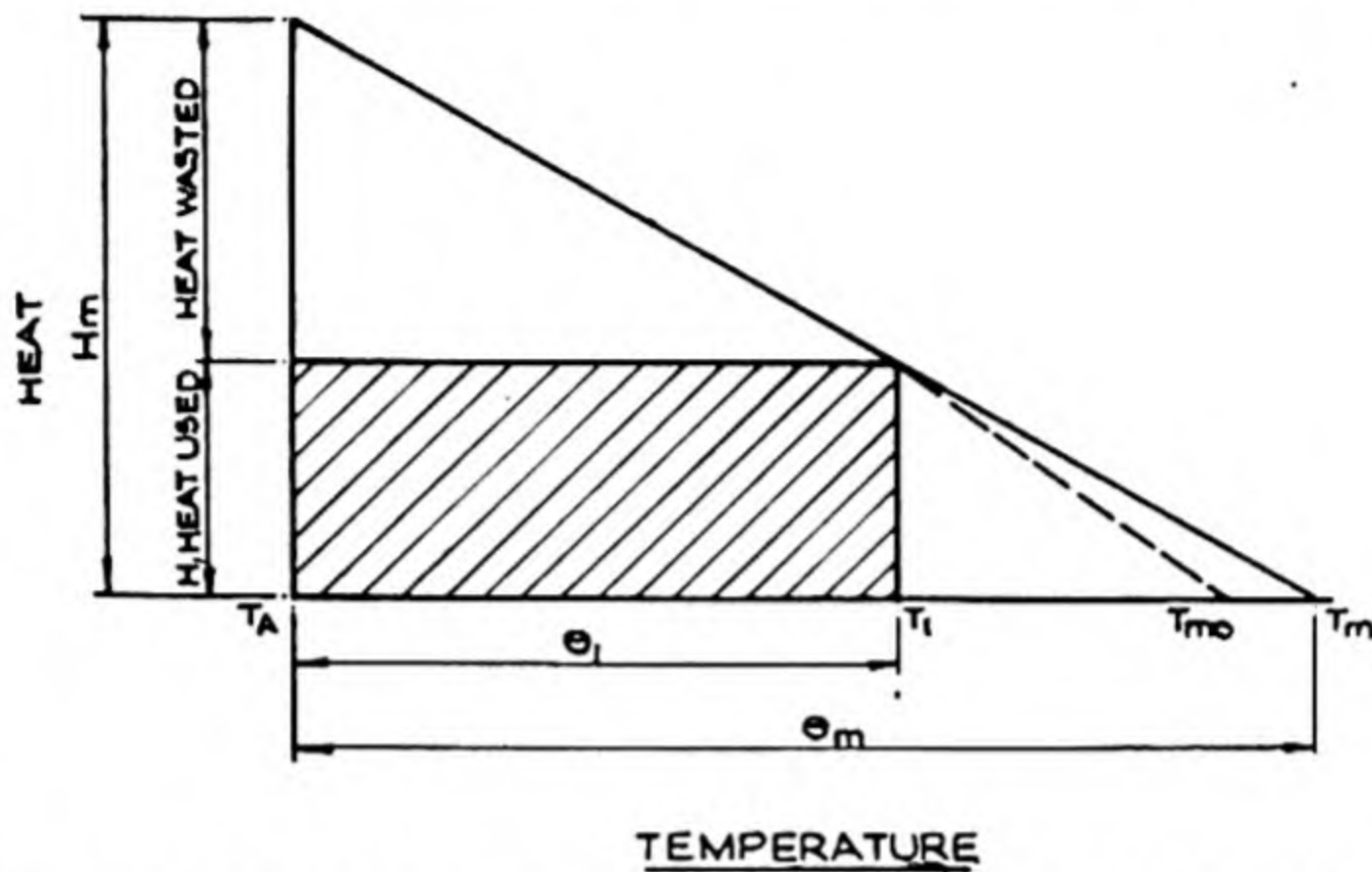


Fig. (2)16. Available Heat in Combustion Gases as a Function of the Temperature of the Body to be Heated.

temperature of combustion is raised from  $\theta_m$  to  $\theta + \eta\theta_1$  ( $T'_m$  in Fig. (2)17), but that *the heat which finally leaves the system and the fuel heat which is required per unit of heat transferred are the same as would be obtained in a system without a preheater in which the*

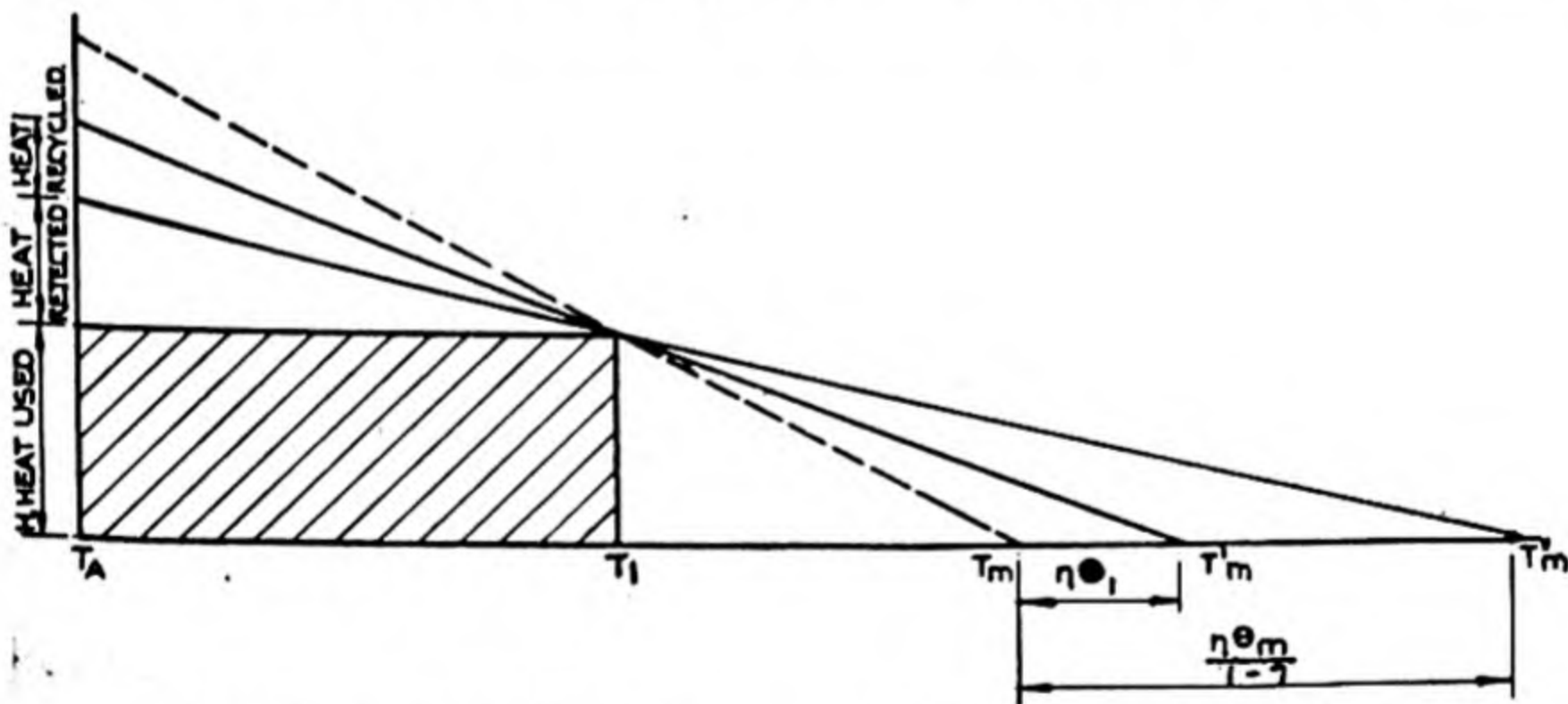


Fig. (2)17. Ideal Heat Temperature Diagram for Preheat.

theoretical temperature was raised from  $\theta_m$  to  $\frac{\theta_m}{1-\eta}$  ( $T''_m$  in Fig. (2)17). This is because the preheater has the double effect of raising the theoretical temperature and also recycling some of the

is shown plotted against their temperature, the dotted line curve corresponding to allowance for dissociation and the straight-line curves for the assumption of the constant specific heat.



heat necessary for this theoretical temperature by extracting from the waste gases. This is a very important point, since an oxygen plant can extract heat from the waste gases to produce the oxygen-enriched air for combustion, but does not have the second advantage of actually increasing the ingoing heat energy. These results are shown diagrammatically in Fig. (2)17.

A good regenerative preheater has a thermal efficiency of 40% so that although not strictly in line with the thermodynamic ideal of making the furnace perfect, it is fair to insert the value  $\eta=0.40$  in this equation. The efficiency is then found to be equivalent to that of a combustion system in which the theoretical maximum temperature is raised from the original value of  $2200^\circ$  to roughly  $3670^\circ \text{C}.$ \*

In the same way the use of oxygen-enriched combustion air may be directly compared with preheat in the case where the power required to operate the oxygen plant is regarded as obtained by waste heat from the heating system. In general, the amount of waste heat is reduced by the use of oxygen so that there is a limit to the application. But even where the oxygen is made with electricity from the grid the analysis in terms of thermodynamics given below is valid since the waste heat can be converted with a fairly high efficiency to steam which is equivalent to the steam used in the power station to produce the electricity by which the oxygen is made. In order to treat the whole system in terms of energy a figure of 300 kWH required to produce 1 tonne of oxygen in a mixture of 90% oxygen concentration by volume has been taken. From this it follows that the work required to produce 1 tonne of oxygen in a mixture containing a volume fraction  $\alpha$  of oxygen is given by

$$392\left(1 - \frac{0.21}{\alpha}\right) \text{kWH} \quad . \quad . \quad . \quad . \quad (2/4)$$

The work necessary to produce the 32 gm. of oxygen which are required to burn 12 gm. of carbon and give a combustion heat of 94.2 kcal. is thus

$$10.8\left(1 - \frac{0.21}{\alpha}\right) \text{kcal.} \quad . \quad . \quad . \quad . \quad (2/5)$$

At this point it is again necessary to introduce a deviation from the ideal. The efficiency of power production *from waste heat* on a

\* This figure corresponds to a point on the  $T$  axis through which lines are drawn and not to an actually realisable temperature.



large scale is of the order of 20%, so that the waste heat required to produce 32 gm. of oxygen is five times this last figure. For example, the power required to produce 32 gm. of oxygen in a 90% mixture is 41.4 kcal., i.e. nearly half that realised by the combustion of the carbon. This emphasises the fact that it is essential to regard the power used to produce the oxygen for combustion as part of the fuel requirements of the system and if it is taken from an external electricity supply it must be regarded as having a coal equivalent of 4-5 times the direct thermal equivalent of the electricity.

The effect of enriching the combustion air, Fig. (2)18, is to reduce the amount of nitrogen in this air and hence provide a direct increase in the combustion temperature. Taking the specific heat of  $\text{CO}_2$  as roughly  $0.57 \text{ kcal./m}^3 \text{ } ^\circ\text{C}$ . and that of nitrogen as

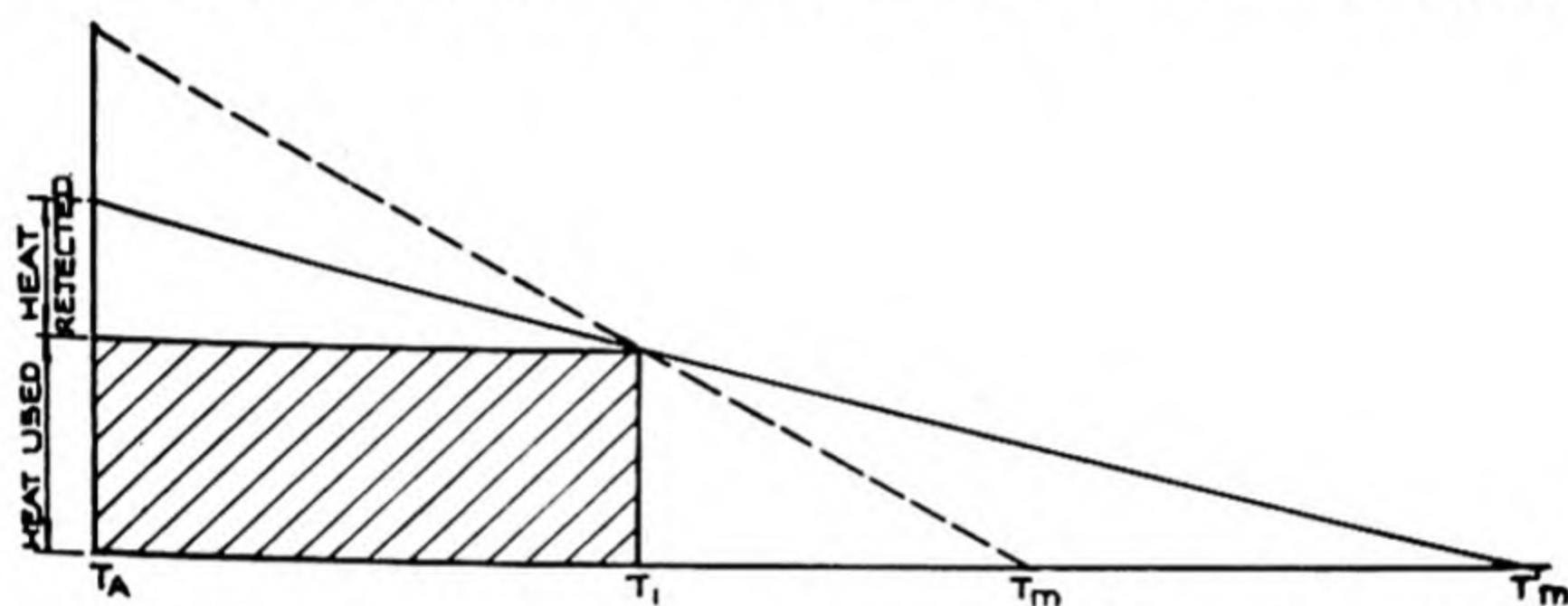


Fig. (2)18. Ideal Heat Temperature Diagram for Combustion with Oxygen Enriched Air.

0.35, the theoretical temperature of combustion of pure carbon  $\theta'_m$  is given by the expression

$$\frac{\theta'_m}{\theta_m} = \frac{1}{0.30 + 0.185 \left( \frac{1}{\alpha} - 1 \right)} \quad \dots \quad (2/6)$$

The various results are summarised in Table 2.5. The numerical consequences of these formulæ are compared for two values of  $\eta$  and two of  $\alpha$  in Table 2.6. The formulæ show that an oxygen concentration of 38% gives the same fuel saving *for all values of*  $\theta_1$  as the 40% efficient preheater. It thus appears that oxygen-enriched combustion is thermodynamically equivalent at all temperatures to that preheat system which has a preheater efficiency  $\eta$  given by the relation

$$1 - \eta = 0.30 + 0.185 \left( \frac{1}{\alpha} - 1 \right) \quad \dots \quad (2/7)$$

where  $\alpha$  is the fractional oxygen concentration in the mixture. In



TABLE 2.5

COMPARISON OF THE PRICE OF OBTAINING  $H_1$  UNITS OF HEAT AT TEMPERATURE  $T_1$  IN VARIOUS WAYS

	No preheat	Preheat	O <sub>2</sub> enrichment (using power produced by waste heat)
Total external heat required	$H_1 \frac{\theta_m}{\theta_m - \theta_1}$	$H_1 \frac{\theta_m}{\theta_m - (1 - \eta)\theta_1}$	$H_1 \frac{\theta'_m}{\theta'_m - \theta_1}$
Heat wasted	$H_1 \frac{\theta_1}{\theta_m - \theta_1}$	$H_1 \frac{\theta_1 \cdot (1 - \eta)}{\theta_m - (1 - \eta)\theta_1}$	$H_1 \frac{\theta_1}{\theta'_m - \theta_1}$
Heat recycled	—	$H_1 \frac{\theta_1 \cdot \eta}{\theta_m - (1 - \eta)\theta_1}$	—
Heat used in O <sub>2</sub> production (20% efficiency)	—	—	$0.58 \frac{H_1 \theta'_m}{\theta_m - \theta_1} \left(1 - \frac{0.21}{\alpha}\right)$
Ideal efficiency	$\frac{\theta_m - \theta_1}{\theta_m}$	$\frac{\theta_m - (1 - \eta)\theta_1}{\theta_m}$	$\frac{\theta'_m - \theta_1}{\theta_m}$

No wall losses, infinite area for heat transfer, instantaneous combustion. Also assume rich fuel burnt with air where  $\theta'_m$ =constant specific heat of gases. No excess air.  
 $\theta_m$  is the theoretical maximum combustion temperature for non-preheated air and fuel.

TABLE 2.6

NUMERICAL VALUE FOR PRICE OF HEAT AT VARIOUS TEMPERATURES

Temperature excess of heated body over surroundings. $\theta_1$	Number of heat units required as fuel to produce one heat unit at $\theta_1$ .				
	No preheat	Preheat		Oxygen	
		$\eta = 0.40$	$\eta = 0.80$	$\alpha = 0.30$	$\alpha = 0.75$
100	1.05	1.03	1.01	1.04	1.02
300	1.16	1.09	1.03	1.12	1.05
500	1.30	1.16	1.05	1.21	1.09
750	1.52	1.26	1.07	1.35	1.14
1000	1.84	1.38		1.53	1.20
1250	2.33	1.52		1.77	1.26
1500	3.16	1.70		2.09	1.33
1650	4.04	1.82		2.34	1.37
	$\frac{1}{1 - \frac{\theta_1}{2200}}$	$\frac{1}{1 - \frac{\theta_1(1 - \eta)}{2200}}$		$\frac{1}{1 - \frac{\theta_1}{\theta'_m}}$	
	Where $\theta'_m = \frac{2200}{0.30 + 0.185\left(\frac{1}{\alpha} - 1\right)}$				

TABLE 2.7  
USES OF OXYGEN FOR COMBUSTION IN THE IRON AND STEEL INDUSTRY

Required oxygen concentration 1	% O <sub>2</sub> in mixture 2	Amount of pure O <sub>2</sub> used, ft. <sup>3</sup> /ton of iron or steel produced 3	Overall increase of output obtained from given furnace, % 4	Fuel used at present/ton iron or steel 5	Percentage fuel saved by O <sub>2</sub> 6	Approx. annual tonnage of metal made by this process in Great Britain 7	References 8
<i>Open Hearth</i> O <sub>2</sub> used during melting Cold metal Hot metal O <sub>2</sub> used during refining: Jet Lance	>90 90 90 90 90 90	870 1610 740 102 160	36 44 30 10 7 25-35 2	29 gals. oil 4.5 × 10 <sup>6</sup> B.Th.U. 4.0 × 10 <sup>6</sup> B.Th.U. 4-5 × 10 <sup>6</sup> B.Th.U. 4-5 × 10 <sup>6</sup> B.Th.U.	26 — 6 7 5 3	11 million	1 4 2 11 4
<i>Blast Furnace</i> Ordinary design: Ferro-silicon alloys Ferro-silicon	32 26 33	20,000 16,000 22,000	100 70 120	32 cwt. coke 46 cwt. coke 46 cwt. coke	25 10 32	Ferro alloys 35,000	3 9
Ordinary iron Low shaft <i>Cupola</i>	24-26 90-100 No experi- ments	4000-8000 21,000	—	20 cwt. coke	10-15 50	8 million tons —	7, 8 6
<i>Bessemer</i> Bottom blown	100	1400	150-250	—	Saving by re- duced re- quirements of molten iron —	0.93 million tons	5
Side blown	90	1500	Not known yet	—	—	Small	10

## References for Table 2.7

- 1 Kopecki, *Iron Age*, 1946, No. 28, 47.
- 2 J. D. Knox, *Steel*, 30 June 1947, 86
- 3 I. P. Bardin, *Bull. Acad. Science, U.R.S.S.*, 1946, 1385-99.
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- 5 V. V. Kondakov, *Bull. Acad. Sciences de L'U.R.S.S.*, 1946, No. 10, 1401-1420; *I.S.I. Trans.*, No. 316, Aug. 1947.
- 6 R. Durrer, *J.I.S.I.*, June 1947, vol. 156, 253. (These figures are calculated only.)
- 7 W. Lennings, *Stahl und Eisen*, 1935, 55, 533-565.
- 8 W. Lennings, *Stahl und Eisen*, 1943, 63, p. 757.
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- 10 John Miles, *Private Communication*.
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(Courtesy of *Inst. Chem. Eng.*)



## 2.2.9 THE SCIENCE OF FLAMES AND FURNACES

practice, preheat efficiencies vary owing to engineering considerations with the temperature they are required to produce ; a figure of 80% ( $\eta=0.80$ ) is obtainable when the heating gas temperature is below  $1000^{\circ}\text{C}$ ., whereas the regenerators used in open-hearth steel and glass furnaces which deal with heating gases in the range  $1400\text{--}1650^{\circ}\text{C}$ . rarely exceed 40%. This is the main reason *why oxygen for combustion is in practice the more attractive the higher the working temperature required.*

At the moment a preheat system is cheaper in capital cost than an oxygen plant, so that oxygen cannot be expected to oust preheat systems where, as in the open-hearth, these are fully established. There are, however, in such cases special considerations which make the use of oxygen as a supplement to preheat of very great interest. In cases such as the Bessemer converter and the cupola, where the use of waste heat to provide preheat has not so far proved practical, an oxygen plant may have very important advantages which will give an overall fuel saving, even though it cannot be produced with power from waste heat. Table 2.7 gives the main results so far obtained in the use of oxygen in steel furnaces.

**2.2.10. Conclusions from the Application of the Second Law of Thermodynamics to Furnace Heating.** The above discussion can be extended to bring out a number of very important principles which result from the application of the second law of thermodynamics to furnace-heating problems. These principles may be enunciated as follows :

(1) Heat losses are of very much greater importance when they take place from a part of the system at a higher temperature than when they take place from a part at a lower temperature. As an example of this principle the case of the open-hearth furnace is interesting. Here the heat losses from the furnace proper are greater than the amount of heat utilised in the bath ; hence, if they could be eliminated the fuel consumption would be reduced to less than one half of its present value. Elimination of an equal amount of heat loss from the regenerator would result in a saving of about one-fifth of the total fuel, while the elimination of an equal amount of heat losses from the gases after they have left the regenerators would result in no saving at all.

A corollary of this principle is that it is very important not to take the heat for preheating air from the hottest part of the system unless the heat taken in this way is merely heat which must be lost



from the hot zone owing to requirements for preservation of refractories, e.g. where the alternative is water cooling or cooling with useless air.

(2) The counterflow system of heating, whether of a gas or a solid, can lead to very much better utilisation of the heat of different qualities than can parallel or cross flow or batch heating.

(3) Air preheat results in a fuel saving greater than that which would correspond to the virtue of the waste heat taken from the gases, because in addition to recycling energy it reduces the loss of virtue on combustion.

(4) Where it is not possible to make use of the heat of all grades in the furnace system itself, either by a counterflow heating cycle or by preheaters, the use of a combined system of furnace for high-grade heating and other appliance (such as a waste heat boiler) for lower-grade heating is the only alternative which can result in good fuel saving.

(5) In cases where it is desired to obtain uniform temperatures over a large volume, it is theoretically preferable to do this by recirculating a certain fraction of the gases leaving a furnace chamber and mixing them with the incoming gases, rather than by increasing the excess air in the former. This is because recirculating the gases makes up for the lowering in the virtue of the heating gases consequent on the dilution, by the virtue brought back by the recirculating gases, whereas with air dilution the extra air brings no virtue with it.

**2.2.11. The Ideal Furnace as Specified by Thermodynamics.** Section 2.2 can be summarised by considering the ideal furnace from the point of view of maximum utilisation of the virtue of the fuel and hence minimum fuel consumption. The nature of the ideal furnace must, however, depend to a certain extent on the actual thermal requirements of the charge. The possible varieties of these can be roughly divided into three classes each of which has its own ideal furnace. They are therefore discussed separately under the headings (a), (b), (c) below.

(a) Material of effectively constant specific heat which can be heated at a constant rate and start to cool, or be removed from the furnace as soon as it reaches the maximum temperature. Examples of such material are bricks and pottery made from raw materials such that the endothermic firing reactions are small compared with the sensible heat, and the reheating of steel billets for rolling. The



## 2.2.11 THE SCIENCE OF FLAMES AND FURNACES

ideal furnace for firing material of this type is one in which heating proceeds by a straightforward counterflow process. The fuel is burnt instantaneously at the ingoing end, and flows down the heating chamber in such a way that its temperature retains about the same excess above that of the charge with which it is in contact at each point. Instantaneous combustion is ideal thermodynamically, as it ensures that the maximum temperature difference for heat transfer is available at all points, although in practice instantaneous combustion is often made impossible by restrictions due to the refractory materials. The heat transfer coefficient between the gases and the charge and the surface of contact must ideally both be very large so that the necessary heat can be forced into the charge with a minimum temperature difference. In this way the gases leaving at the cool end of the furnace would be brought down to about  $200^{\circ}\text{C}$ . so that the heat of each quality is utilised to a maximum extent. If a large contact area between the gases and the charge is rendered impossible by the necessity to avoid chemical interaction, then the gases will have to leave the furnace at a considerably higher temperature in spite of the counterflow principle, and it will be necessary to have a metallic recuperator to use the lower-grade heat from the gases.

*(b) Material of practically uniform specific heat which has to be maintained at a certain temperature for a definite period while it is being processed.* The best known examples of this case are steel and glass. In this case the furnace must consist of a constant-temperature chamber directly heated by the combustion gases, together with a continuous counterflow chamber or a series of chambers for intermittent counterflow, which are heated by the gases leaving the primary chamber. The temperature in the primary chamber must be maintained uniform over its whole floor area either (1) by firing the gases across the shorter dimension, or (2) by recycling a quantity of the gases leaving the chamber which are mixed in with the hottest gases entering, or (3) if neither of these measures is possible, by controlled delayed combustion. The ideal method of carrying out controlled delayed combustion, i.e. combustion in which not all the fuel energy is released before it has been appreciably cooled by heat transfer to the charge, is the delivery of a certain fraction of the combustion air at each of a number of points along the chamber in such a way that there is instantaneous combustion at each point. Where delayed combustion is obtained by relying on diffusion between air and gas it



will always be necessary to have high excess air to avoid incomplete combustion (see 3.3 below). In furnaces for heating this type of material there is bound to be more heat in the gas leaving the first chamber than can be utilised in the subsequent counterflow chambers. This follows because although the first chamber does not supply any useful heat to the material, there will be heat losses in it during the whole of the working period which must be made up. This shows also that insulation of the first chamber is of especial importance, since if the heat losses from this chamber are very great the subsequent counterflow chambers will not absorb an appreciable fraction of the heat in the gases leaving the first one. In any case, a recuperator is bound to be necessary in furnaces of this type, but if the counterflow chambers are well utilised and the first one well insulated, the gases may leave not above  $600^{\circ}$  or  $700^{\circ}$  C., so that the recuperator can make good use of their remaining energy.

(c) *Material where the bulk of the heat must be supplied at the maximum temperature.* A typical example is the formation of calcium carbide from lime and carbon, and carborundum from sand and carbon. Even in these cases the ideal furnace would be one in which the incoming materials were heated by counter-

flow, simply in order that the main chamber shall not be burdened by this extra task and in order to drop the temperature of the gases

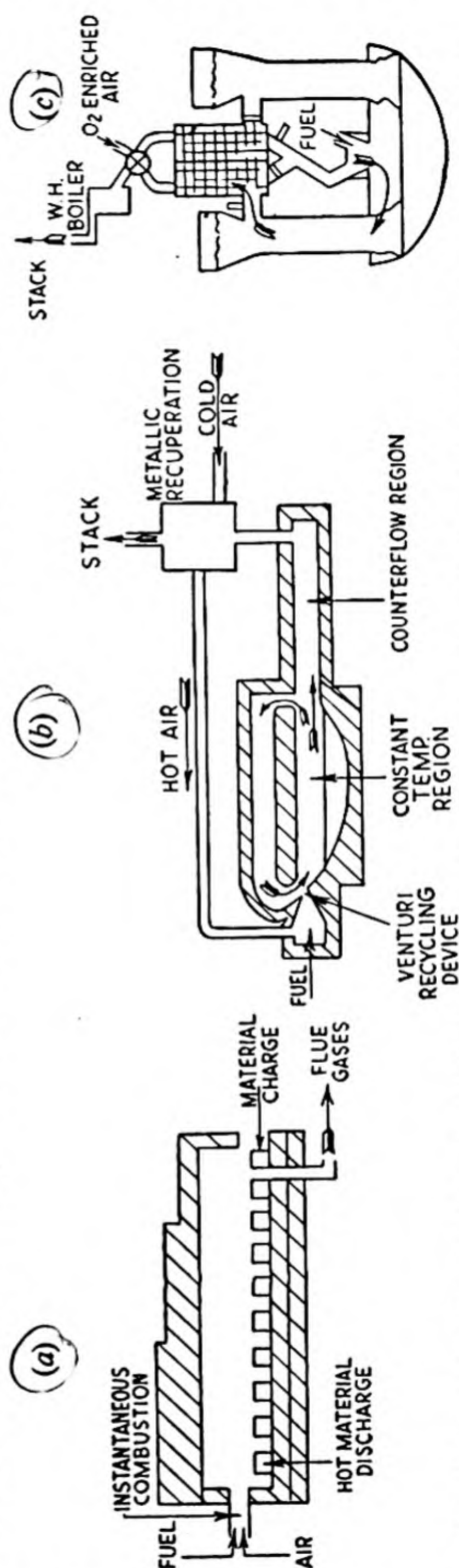


Fig. (2)19. Ideal Furnaces for Material which (a) has merely to be heated; (b) has to be heated and maintained at Constant Temperature; (c) has High Heat Absorption at the Maximum Temperature.



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leaving the main chamber to a value slightly below the temperature of this chamber. It follows, however, that the bulk of the heat in these gases must be used in a large heat exchanger, which must almost certainly in this case be of the regenerative type, since there will be a large amount of liquid material in the gases. In this case also the use of oxygen-enriched air is of especial importance, to bring as much of the heat of the gases above the working temperature as possible. It is possible that the regenerators will still not be able to make use of the whole of the heat in the gases owing to the greater heat capacity of the waste gases compared with the fuel and air and to limitations on the heat transfer coefficients, and hence in the ideal system these gases will pass to a waste-heat boiler or other independent appliance after leaving the heat exchanger.

These three types of furnace are shown in Fig. (2)19.

### 3. THE LIBERATION OF HEAT BY COMBUSTION

#### 3.1. The Energy Release and Equilibria of Chemical Reactions

**3.1.1. Introduction.** As the heating of all furnaces discussed in this monograph is carried out as a result of chemical reactions, it is of great importance to know as much as possible about these reactions. There are three kinds of knowledge required, namely :

- (i) What is the energy released by the reactions when they proceed to completion ?
- (ii) How far and in which direction can the reactions be expected to proceed given unlimited time and space, but finite and known heat loss to surroundings ?
- (iii) How far will the reactions proceed with given limitations of time and space ?

The answers to the first two of these questions can be given with a high degree of reliability as a result of the theoretical and practical work of several generations of scientists. A very brief summary of these answers will be given in the first section of this Chapter (3.1.2 and 3.1.3), while the next four sections (3.2–3.5) deal with the third question as to the speed of combustion, the last section (3.6) being concerned with the control of combustion conditions.

In designing the arrangements for combustion in a furnace the most usual aim is to obtain approximately complete combustion in a desired space with none, or a minimum, of excess air. To give the highest temperature difference for heat transfer this desired space should, as discussed in the previous chapter, be a very small fraction of the furnace chamber in which heat transfer takes place. In certain cases, however, it must be a relatively large fraction of the furnace chamber. Two examples are :

- (1) When the heat transfer coefficient is higher with delayed combustion, as with luminous radiation, or
- (2) when a very short flame gives excessive local heating, as might occur, for example, if the combustion of an open-hearth furnace flame was complete in one-quarter or less of the furnace length.



### 3.1.1 THE SCIENCE OF FLAMES AND FURNACES

The knowledge of the factors governing the rate of chemical reaction between fuel and air or oxygen is therefore necessary not only to calculate how a given arrangement will work but also to predict what types of alterations could give more complete combustion with less excess air.

The subject of combustion shows in a very fascinating way the possibilities for the fusion of pure scientific research and the practical design of industrial appliances. Thus, for example, as will be seen in Section 3.2.1, it is necessary to appeal to the subject of wave mechanics in order to understand the mechanism of any chemical reaction. The ignition of mixed gases entering a furnace depends almost entirely on the mechanism of such a chemical reaction, so that formulæ which are of direct value to the furnace designer have their roots in wave mechanics.

It must, however, be pointed out that even on the academic side research into combustion has so far tended to be more fruitful for the needs of the designer of the internal combustion engine and the gas turbine rather than for those of the furnace designer. The value to the designers of these engines of the work which has been carried out is extremely evident and there is no doubt that a sufficient drive on the part of the furnace industry would have repercussions both in the Proceedings of the Royal Society and in the practical advances which could come in furnace design. This point is illustrated by the book of Lewis and von Elbe,<sup>3.1</sup> where the applications of the knowledge of combustion obtained in the laboratory to industrial heating take two pages, while those to the internal combustion engine take nearly thirty.

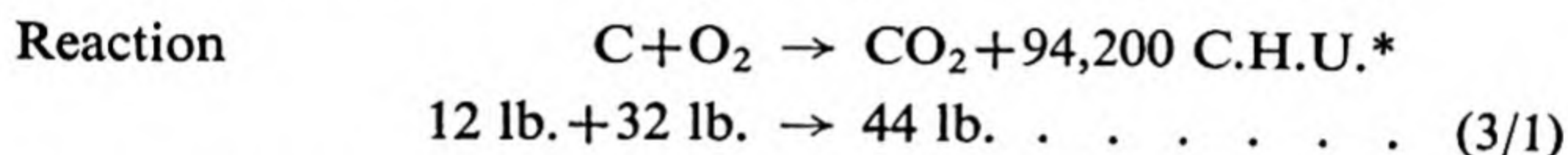
It is true that the great difference in the impact of modern reaction kinetics on the internal combustion engine compared to that on the furnace industries may be partly attributed to the fact that the latter are concerned with two-phase and diffusion combustion reactions which depend on physical processes to a very large extent, especially as much more heat is available for ignition in furnaces. Nevertheless, there is a clearly sustained connection between the former industry and the fundamental scientist which is lacking in the case of furnaces, and which would undoubtedly lead to many fruitful results if it could be extended to them. The gas-turbine engine to some extent provides a link between the application of fundamental work on reaction kinetics to the internal combustion engine and to the industrial furnace, because on the one hand it

<sup>3.1</sup> LEWIS, B., and VON ELBE, G.: *Combustion Flames and Explosion of Gases*.

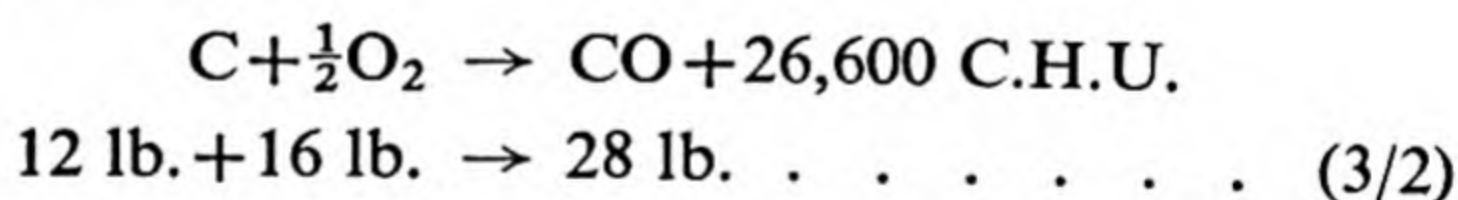


lends itself to laboratory studies more readily than the industrial furnace and on the other it depends on aerodynamic mixing as well as chemical reaction rates.

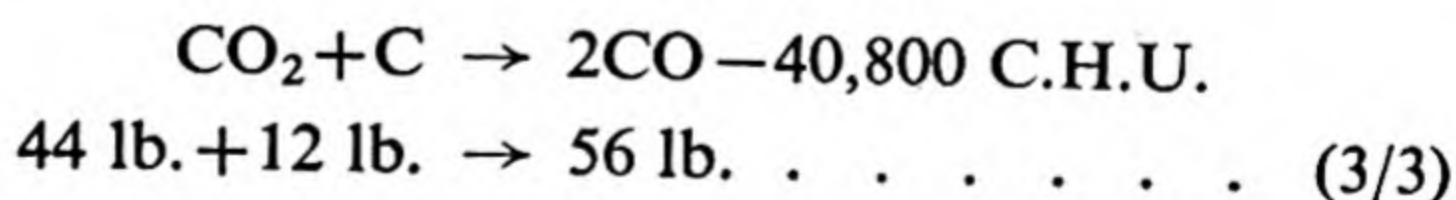
**3.1.2. Reaction Energies.** In this section a very short discussion of the reactions which are most important from the point of view of fuel combustion will be given. In furnace firing most chemical reactions take place at atmospheric pressure, so that heats of reaction at constant pressure may be used for calculation. The energies of reactions are given for the weights of reacting materials equal to the molecular weight of each species in lb. multiplied by the number of molecules of that species reacting. The corresponding weight is written underneath in each case. In the case of gases the molecular weight of the gas in lb. occupies 359 ft.<sup>3</sup>, so that 32 lb. of oxygen (O<sub>2</sub>) or 2 lb. of hydrogen (H<sub>2</sub>) or 44 lb. of carbon dioxide (CO<sub>2</sub>) occupy this volume. When air is used for combustion 32 lb. of O<sub>2</sub> is accompanied by approximately 106 lb. of N<sub>2</sub> and the mixture occupies approximately 1710 ft.<sup>3</sup>.



The aim of the designer who wishes to fire a furnace directly with coal is to make this reaction proceed to completion as rapidly as possible in order to liberate all the chemical energy of the carbon in the useful furnace space.



This reaction, carried out in one stage, would represent the process in an ideal dry gas-producer, and the fact that there is still a considerable heat release is the reason why a dry-blast gas-producer must give hot gas and cannot have a "cold-gas efficiency" (first law) above 80%.† The other two reactions :



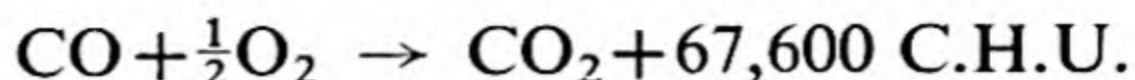
\* *Units.* It should be noted that if the weights of reactants and products are expressed in kilos, the figures for the heats of the reactions will be in kilocalories instead of C.H.U.s.

† It will be shown later that none of these reactions proceed to completion and since the  $\text{CO}/\text{CO}_2$  ratio increases with temperature, the ideal gas producer must run at a high temperature and the ideal boiler fuel bed to produce  $\text{CO}_2$  must run at a low temperature.



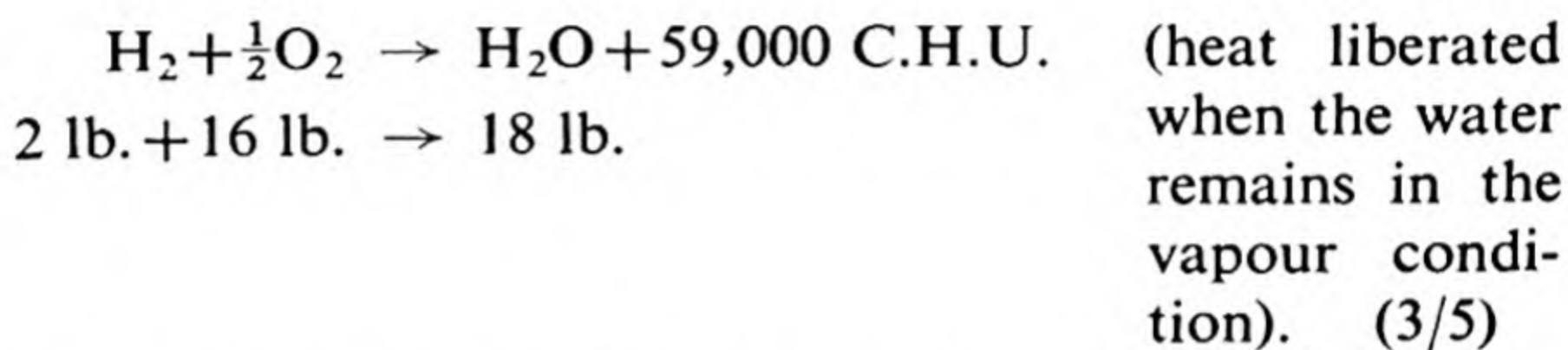
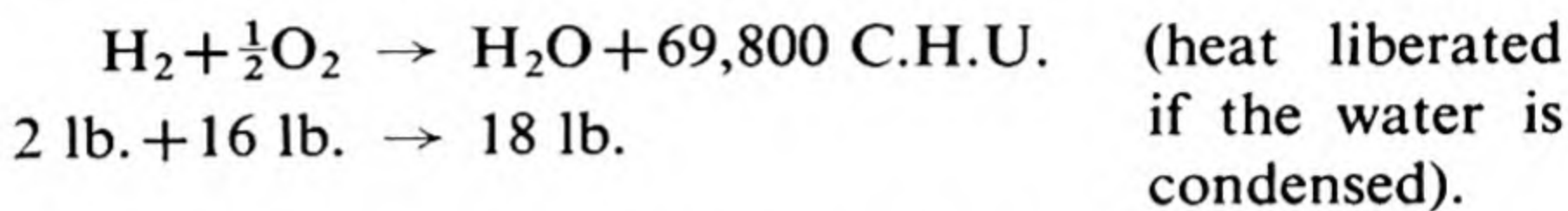
### 3.1.2

and



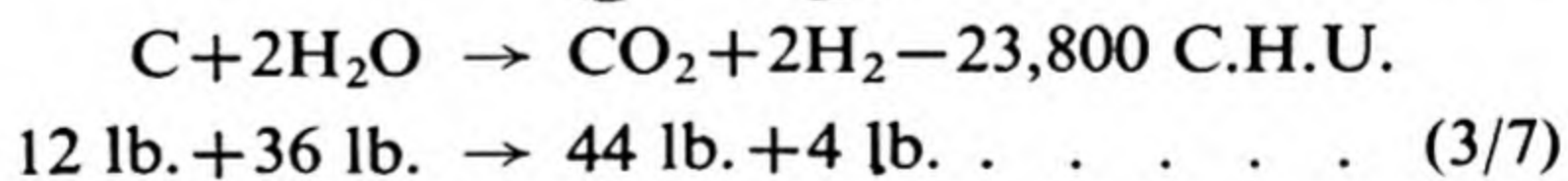
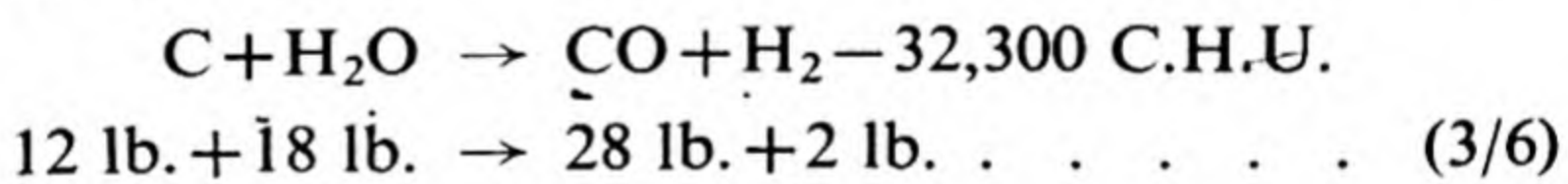
can be deduced from these first two and correspond respectively to the reduction of  $\text{CO}_2$  in the latter part of the bed of the gas-producer, and to the combustion of  $\text{CO}$  in the producer gas flame or the combustion with secondary air over the surface of a deep fuel bed.

The reaction of hydrogen with oxygen, expressed by the formula :



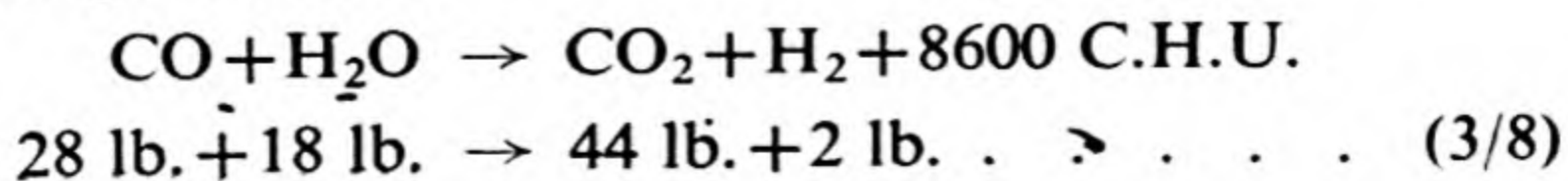
takes place in the combustion of producer gas made with steam-air mixture and of the products of distillation from coal over a surface of a fuel bed.

In the three reactions between water vapour and carbon oxides, water vapour is to some extent analogous to  $\text{CO}_2$  in the first two, hydrogen being equivalent to CO. These two reactions: -



represent the decomposition of steam by carbon under conditions where heat is continually supplied, the former predominating at higher temperatures and the latter at lower temperatures. This heat may be supplied by absorption of sensible heat from the gases, or from the carbon, which have been previously heated to a very high temperature, or by heat transfer from heated surrounding walls.

The remaining reaction :





illustrates the equivalence between CO and  $H_2$  as fuels, since the energy change is relatively small. These last three reactions (3/6–8) are of importance in gas-producers to which steam is added and in water-gas generators, and the third also occurs to some extent when producer gas or mixed gas is preheated in an open-hearth regenerator.

The heats of reaction corresponding to these equations are given for reactants starting at a temperature of  $25^\circ C.$  and products finishing at this temperature and with the reaction carried out at atmospheric pressure. As stated above the pressure in furnace systems is nearly always sufficiently close to atmospheric for all pressure corrections to be neglected. On the other hand, the effect of temperature is quite large, especially in reactions where the volume of combustion products is not the same as that of the materials. It follows from the first law of thermodynamics that the heat of reaction at a temperature  $T$  is equal to the heat of reaction at  $25^\circ C.$  plus the difference between (1) the heat required to raise the reacting substances from  $25^\circ C.$  to the temperature  $T$  and (2) the heat required to raise the products of the reaction over this same temperature range. This is because the total energy change must be the same whether the gases react at  $25^\circ C.$  and are then heated to  $T^\circ C.$ , or are heated to  $T^\circ C.$  and then react. Where, however, reaction energies are expressed at  $25^\circ C.$  and the first law is applied with all sensible heats calculated from this temperature, this correction is automatically made, e.g. if air and gas initially at  $25^\circ C.$  are burnt adiabatically the sensible heat of the combustion products above  $25^\circ C.$  will be equal to the heat of reaction at  $25^\circ C.$

The following formulæ are derived from the basic chemical equations 3/1–8 and are given here because they are useful in practical combustion calculations. (For a fuller discussion of this subject reference should be made to Haslam and Russell<sup>3.2</sup> or to the Institute of Fuel *Symposium on Waste Heat Recovery from Industrial Furnaces*.<sup>3.3</sup>)

(1) Given the composition by weight (ultimate analysis) of a solid or liquid fuel: required to calculate the theoretical air and combustion gas volumes, the composition of the ideal combustion gases and the excess air ratio (in this case the  $CO_2$  or  $O_2$  of the

<sup>3.2</sup> HASLAM, R. T., and RUSSELL, R. P.: *Fuels and Their Combustion*, 10, p. 193 (McGraw Hill).

<sup>3.3</sup> Inst. Fuel, *Symposium on Waste Heat Recovery from Industrial Furnaces* (Chapman and Hall, 1948).



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actual combustion gases being also given). If the symbols C, H<sub>2</sub>, H<sub>2</sub>O, O<sub>2</sub>, N<sub>2</sub> and S are used to represent the fractional weights of these elements in lb. in 1 lb. of fuel as fired, then :

Theoretical volume ( $A_o$ ) of air per lb. of fuel

$$=1710\left(\frac{C}{12}+\frac{H_2}{4}-\frac{O_2}{32}+\frac{S}{32}\right)n \text{ ft.}^3/\text{lb.} \quad . \quad . \quad (3/9)$$

Theoretical volume ( $V_o$ ) of combustion gases (wet) per lb. of fuel

$$=359\left(4.76\left\{\frac{C}{12}+\frac{S}{32}\right\}+5.76\frac{H_2}{4}+\frac{H_2O}{18}-3.76\frac{O_2}{32}+\frac{N_2}{28}\right)n \text{ ft.}^3/\text{lb.} \quad . \quad . \quad . \quad (3/10)$$

Percentage CO<sub>2</sub> in wet combustion gases =  $300 \frac{C}{V_o} \%$ . In the case where S, H<sub>2</sub>O, O<sub>2</sub> and N<sub>2</sub> are neglected this can be simplified to

$$\frac{2.1 \cdot C}{C+3.63H_2} \quad . \quad . \quad . \quad . \quad . \quad (3/11)$$

Percentage H<sub>2</sub>O in wet combustion gases =  $\frac{359}{V_o} \left( \frac{H_2}{2} + \frac{H_2O}{18} \right) \%$ .

In the simple case this can likewise be simplified to

$$\frac{1.26 \cdot H_2}{C+3.63 \cdot H_2} \quad . \quad . \quad . \quad . \quad . \quad (3/12)$$

Theoretical volume ( $V'_o$ ) of combustion gases (dry) from 1 lb. of fuel

$$=359\left(4.76\left[\frac{C}{12}+\frac{S}{32}\right]+3.76\frac{H_2}{4}-3.76\frac{O_2}{32}+\frac{N_2}{28}\right) \quad . \quad (3/13)$$

Percentage CO<sub>2</sub> in theoretical dry combustion gases (CO'<sub>2</sub>)

$$=300\frac{C}{V'_o} \quad . \quad . \quad . \quad . \quad . \quad (3/14)$$

If the percentage CO<sub>2</sub> in the actual dry combustion gases is measured to have a value which will be expressed by the symbol CO<sub>2</sub> then the ratio ( $n$ ) of excess air to theoretical air is given by

$$n=\frac{A-A_o}{A_o}=\frac{V'_o}{A_o}\left(\frac{CO'_2}{CO_2}-1\right) \quad . \quad . \quad . \quad (3/15)$$

( $A$  = actual air supplied  $n$  ft.<sup>3</sup>/lb. of fuel.)

If the percentage  $O_2$  in the actual dry combustion gases is measured to have a value expressed by the symbol  $O_2$ , then

$$n = \frac{V'_o}{A_o} \frac{O_2}{0.21 - O_2} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (3/16)$$

(2) Given the composition by volume of a gaseous fuel: required to calculate the same quantities as in (1). If the symbols  $CO_2$ ,  $CH_4$ ,  $C_n$ ,  $H_m$ , etc., represent the fractions of these gases present in 1  $n$  ft.<sup>3</sup> of wet flue gas, then:

Theoretical volume ( $A_o$ ) of air to burn 1  $n$  ft.<sup>3</sup> of wet flue gas is given by

$$A_o = 4.76(3.67C_nH_m + 0.5CO + 2CH_4 + 0.5H_2)n \text{ ft.}^3/n \text{ ft.}^3 \quad \cdot \quad (3/17)$$

Theoretical volume ( $V_o$ ) of wet combustion gas from 1  $n$  ft.<sup>3</sup> of wet flue gas

$$V_o = \alpha + \beta + \gamma n \text{ ft.}^3/n \text{ ft.}^3 \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (3/18)$$

where  $\alpha = CO_2 + 2.45C_nH_m + CO + CH_4$  is the volume of  $CO_2$  in the wet combustion gases  $n$  ft.<sup>3</sup>/ $n$  ft.<sup>3</sup> from 1  $n$  ft.<sup>3</sup> of wet flue gas.

$\beta = 2.45C_nH_m + 2CH_4 + H_2 + H_2O$  is the volume of  $H_2O$  in the wet combustion gases from 1  $n$  ft.<sup>3</sup> of wet flue gas.

$\gamma = N_2 + \frac{3.76}{4.76} A_o$  is the fractional volume of  $N_2$  in the wet combustion gases from 1  $n$  ft.<sup>3</sup> of wet fuel gas.

The theoretical volume of dry combustion gas  $V'_o = \alpha + \gamma$ .

(3) Given the calorific value of the fuel (solid, liquid or gaseous). Rosin<sup>3.4</sup> has shown that the combustion properties of all normal commercial fuels can be fairly accurately specified by two parameters, namely, the calorific value and the type of fuel, so that approximate values of the theoretical combustion air and combustion gas volume can be obtained when only the calorific value is known. The appropriate figures based on his work are given in Table 3.1.

<sup>3.4</sup> ROSIN, P. O.: "Total, Recoverable and Returnable Heat in Combustion Gases," Inst. Fuel, *Symposium on Waste-Heat Recovery from Industrial Furnaces* (Chapman and Hall, 1948).



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TABLE 3.1

THEORETICAL AIR REQUIREMENTS ( $A_0$  and  $a_0$ ) AND COMBUSTION GAS VOLUMES ( $V_0$  and  $v_0$ ) FOR VARIOUS FUELS

Type of Fuel	Net C.V., C.H.U./lb.	$A_0$ , ft. <sup>3</sup> /lb.	$V_0$ , ft. <sup>3</sup> /lb.
Solid fuel	3000	57	69
	4000	73	83
	5000	89	97
	6000	106	110
	7000	122	126
	8000	138	142
Oil	9600	163	171
	Net C.V., C.H.U./n ft. <sup>3</sup> (dry)	$a_0$ , n ft. <sup>3</sup> /n ft. <sup>3</sup> (dry)	$v_0$ , n ft. <sup>3</sup> /n ft. <sup>3</sup> (dry)
Producer gas	70	0.98	1.83
	75	1.05	1.88
	80	1.12	1.93
	85	1.19	1.98
	90	1.26	2.03
Coke-oven gas	240	3.94	4.51
	250	4.11	4.73
	260	4.29	4.95
	270	4.46	5.16
Blast-furnace gas	60	0.76	1.61
	63	0.83	1.66
	70	0.89	1.71
Mixed coke-oven and blast-furnace gas	90	1.38	2.21
	115	1.81	2.63
	140	2.23	3.04
	165	2.66	3.46

**3.1.3. Reaction Equilibria.** At any given temperature and pressure a reaction will not proceed completely in either direction, but will result in a certain mixture of the compounds on either side of the equation. However, in the case of the combustion equations 3/1 and 3/8 the equilibrium even at temperatures as high as 2000° C. is very far in the direction of complete combustion, i.e. the percentage of CO or O<sub>2</sub> which remains in a theoretical mixture is very small, but there is a slight dissociation of CO<sub>2</sub> into CO and O<sub>2</sub> and of H<sub>2</sub>O into H<sub>2</sub> and O<sub>2</sub> which results in what may be regarded, as discussed in 2.1.2b, as an additional component of the specific heat of the products of combustion at higher temperatures, since the dissociated molecules combine on cooling and release their reaction energy as sensible heat. Carbon monoxide is not appreciably



dissociated even at  $3000^{\circ}\text{C}$ ., a factor which is of importance in regard to the high temperatures attained with the oxy-acetylene flame. The reaction of  $\text{CO}_2$  with excess carbon and the reaction between steam and carbon monoxide both have, however, equilibria which set a theoretical limit to the gases which can be obtained from gas-producers and water-gas generators.\* The reversible decomposition of  $\text{CO}$  into  $\text{CO}_2$  and  $\text{C}$  at low temperatures ( $400^{\circ}\text{C}$ .) is also of importance, as it results in the deposition of carbon in refractories with their subsequent disintegration.

Although it is one stage nearer to reality than the assumption that all reactions proceed to completion in one direction, the study of reaction equilibria represents only a limiting case of what in fact happens, since (except in the case of the reversible reaction  $\text{H}_2\text{O} + \text{CO} \rightleftharpoons \text{CO}_2 + \text{H}_2$ ) the approach to equilibrium is in many cases slow relative to the contact time available in the furnace system. It is, however, generally possible to see from which side the equilibrium composition will be approached and hence this composition gives a limiting condition for the process at any given temperature. It is accordingly frequently useful to be able to calculate the equilibrium composition at a given temperature.

The law of chemical equilibrium may be set out as follows. The first step is the derivation from the latter of the law of mass action.<sup>3.5</sup> This states that if  $n$  molecules of  $A$  and  $m$  molecules of  $B$  react to give  $p$  molecules of  $C$  and  $q$  of  $D$ , i.e. the reaction formula is



then the reaction rate in the direction from left to right will be given by  $k_1(A)^n(B)^m$  and that in the direction from right to left by  $k_2(C)^p(D)^q$ , where  $A, B, C, D$  refer to the concentration of the corresponding molecules present in the gaseous mixture. By applying the gas laws, these concentrations can be replaced by the partial pressures of gaseous reactants in atmospheres while the concentrations of pure liquids and solids are taken as unity.† In

\* The practical limit is set much more by reaction speeds, discussed below in 3.2–3.5.

† This procedure is permissible because the effective partial pressures of the solids and liquids are their vapour pressures and these remain constant during the reaction at constant temperature, so that putting them equal to unity is equivalent to replacing  $K$  by a modified  $K$ , which can equally well be treated as an equilibrium constant.

<sup>3.5</sup> The rigorous derivation of the law of mass action from the second law of thermodynamics is given in ROBERTS, J. K.: *Heat and Thermodynamics*, 3rd edn., pp. 334–337.



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words, the law of mass action states that the rate at which the reaction proceeds in one direction is proportional to the product of the concentrations of the reacting species each raised to the power of the number of molecules at this species which takes part in the reaction.

At equilibrium, the reaction rates in the two directions must be equal, so that

$$\frac{(A)^n(B)^m}{(C)^p(D)^q} = \frac{k_2}{k_1} = \frac{1}{K} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (3/20)$$

where  $K$  is called the equilibrium constant and is a function of the temperature and pressure of the reactants and of the heat and entropy changes of the reaction.

In calculating the equilibrium condition of such a reaction the initial quantities of  $A$  and  $B$  are given and it is desired to calculate the quantities of  $A$ ,  $B$ ,  $C$  and  $D$  which will be present when equilibrium is attained at any given temperature. Since  $C$  and  $D$  are formed by the known reaction (3/19), there is only one unknown, viz. the number  $N$  of moles of  $A$  which will have reacted with  $m/n$   $N$  moles of  $B$  to give  $p/n$   $N$  moles of  $C$  and  $q/n$   $N$  moles of  $D$ . Hence one extra equation is sufficient to calculate all four quantities, and if  $K$  is known, this equation is provided by 3/20.

Now, if  $K$  is measured at one temperature and the heat of reaction is known as a function of temperature, it is possible to calculate  $K$  at any other temperature by means of a relation derived purely from the application of the second law of thermodynamics. Since at equilibrium a small amount of reaction can proceed reversibly, the change of entropy when such a small amount of reaction takes place at constant pressure and temperature is equal to  $\Delta H/T$ , where  $\Delta H$  is the heat of reaction under these conditions ; from this it can be shown<sup>3.5</sup> that the variation of  $K$  with temperature is given by the equation

$$\left( \frac{d(\log K)}{dT} \right)_p = \frac{\Delta H}{RT^2} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (3/21)$$

where  $\Delta H$  is the heat produced by reaction at constant pressure and  $T$  is the absolute temperature ( $^{\circ}\text{A}$ ).

This equation connecting the equilibrium constant with the heat of reaction and the temperature is known as the van't Hoff isochore of reaction.

<sup>3.5</sup> ROBERTS, J. K. : *Heat and Thermodynamics*, 3rd edn., p. 338.



It will be seen from equation 3/21 that if  $\Delta H$  is positive an increase of temperature increases  $K$ , i.e. pushes the equilibrium to the left (increasing the amounts of  $A$  and  $B$ ). Hence heat is absorbed and the tendency to increase the temperature is opposed. This is an example of the principle of le Chatelier: "a change of one of the factors of an equilibrium causes a rearrangement of the system in such a direction that the factor in question tends to experience a change in the opposite sense from the original change." This principle is of great importance. It implies, for example, that the effective specific heat of a gas in which there can occur a reversible reaction (such as the water-gas reaction or the dissociation of  $\text{CO}_2$ ) will be abnormally *high*, since if it is heated more energy will be needed than would correspond to the normal specific heat owing to the absorption of heat by the small change in composition, and if it is cooled more energy will be given out by the reverse change in composition.

The values of  $K$  for the reactions discussed in the preceding section are shown in Figs. (3)1a and b, and the validity of equation 3/21 connecting them is shown in Fig. (3)2, from which it will be seen that  $\log K$  plotted against  $1/T$  is practically linear in all cases. Fig. (3)3 shows the equilibrium mixtures of  $\text{CO}$  and  $\text{CO}_2$  in the presence of carbon, calculated from the equilibrium constants. It will be seen that at temperatures above  $900^\circ \text{C}$ . the mixture is almost pure  $\text{CO}$ , while below  $450^\circ \text{C}$ . it is almost pure  $\text{CO}_2$ .

A very convenient graphical method for assessing by inspection the way a given oxidation or reduction will go by applying this thermodynamic relationship was given by Ellingham<sup>3.6</sup> and has been further worked out by Richardson and Jeffes.<sup>3.7</sup> Either by integration of equation 3/21 with respect to temperature, by making use of the fact that the variation of  $\Delta H$  with temperature is the difference in specific and latent heats of the reactants and the reaction products, or directly from the application of the second law of thermodynamics to the reversible fractional reaction at constant pressure, it can be shown that

$$-RT \log_e K = \Delta H - T\Delta S \quad . \quad . \quad . \quad (3/22)$$

<sup>3.6</sup> ELLINGHAM, H. T.: "Reducibility of Oxides and Sulphides in Metallurgical Processes," *J. Soc. Chemical Industry*, 63, p. 125 (1944).

<sup>3.7</sup> RICHARDSON, F. D., and JEFFES, J. H.: (i) "The Thermodynamics of Substances of Interest in Iron and Steel Making," *J.I.S.I.*, p. 261 (Nov. 1948). (ii) "The Thermodynamic Background of Iron and Steel Making Processes. The Blast Furnace," *J.I.S.I.*, p. 397, esp. Fig. 5, p. 402 (Dec. 1949).



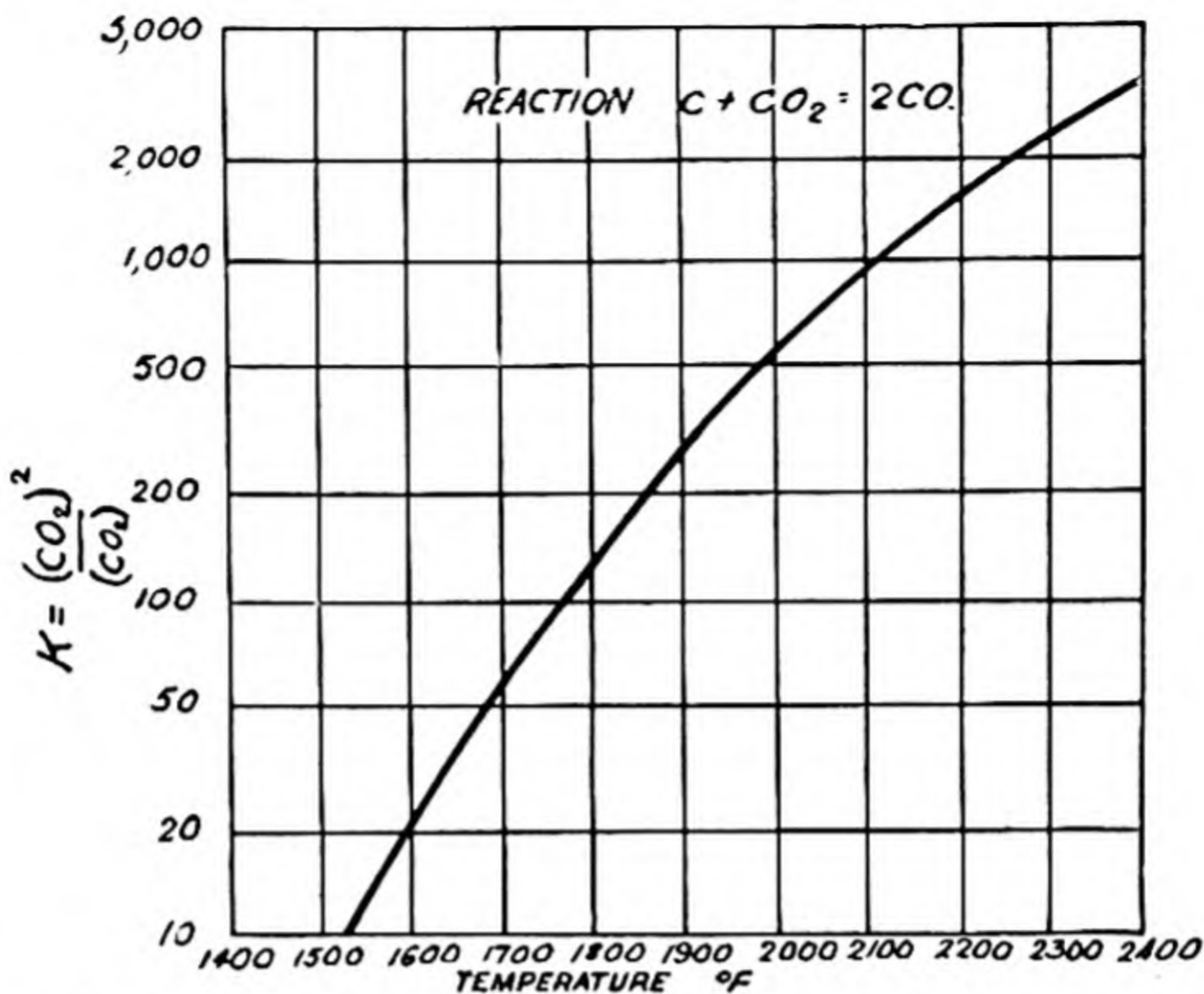


Fig. (3)1. (a) Equilibrium Constant/Temperature Curve for Carbon Reaction.

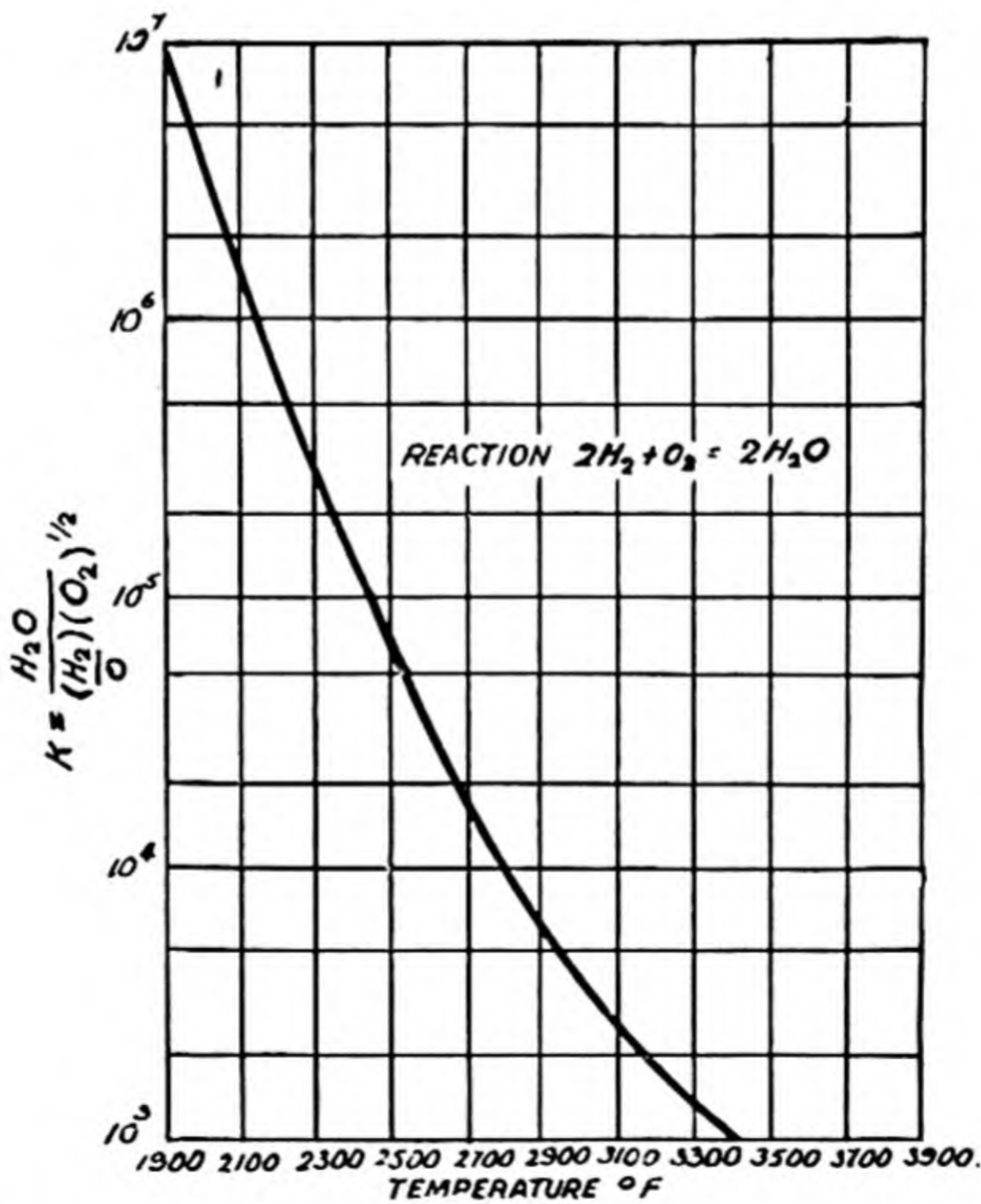


Fig. (3)1. (b) Equilibrium Constant/Temperature Curve for Hydrogen Reaction.

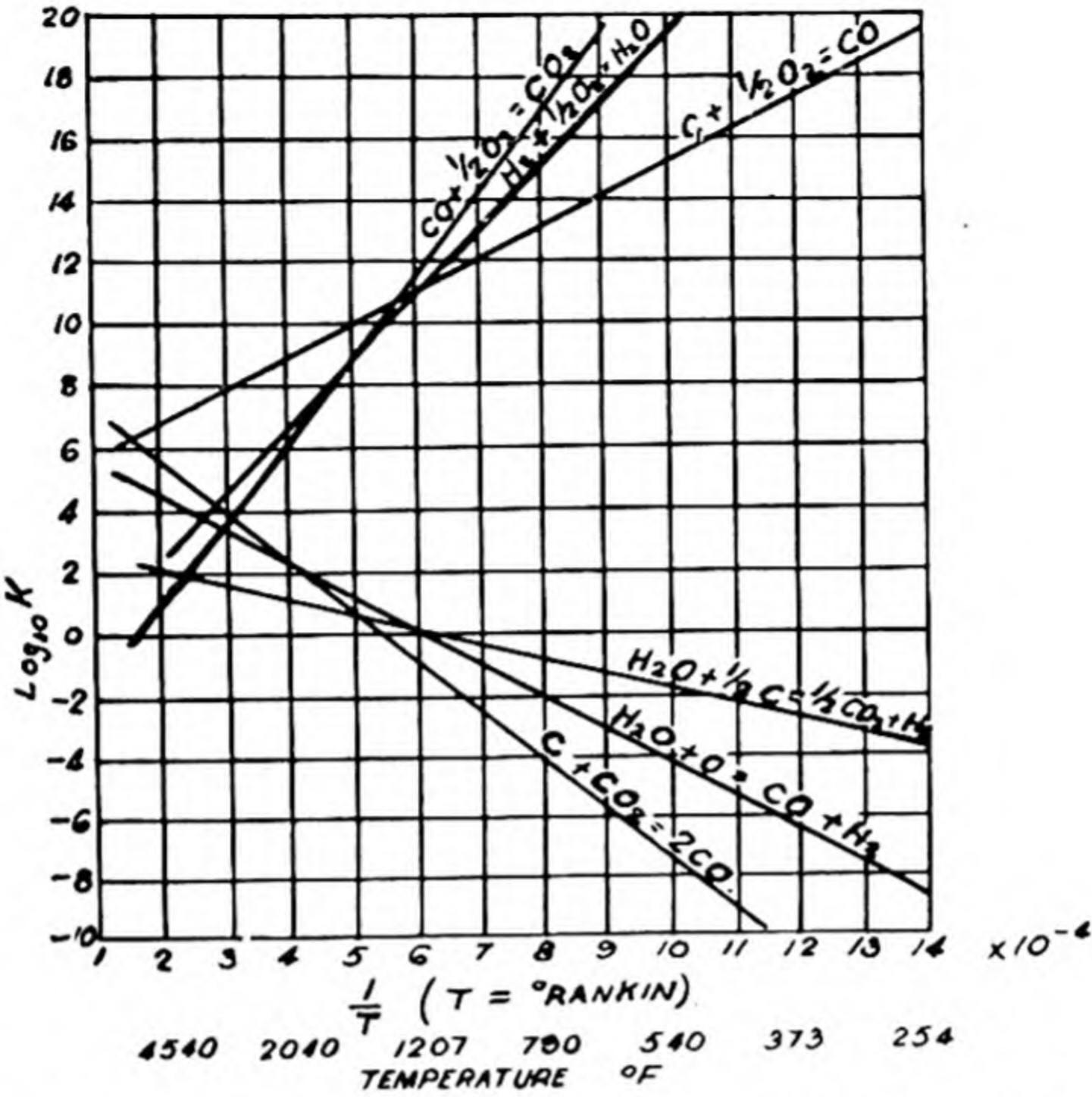


Fig. (3)2. Log K-I/T Curves for Carbon and Hydrogen Reactions.

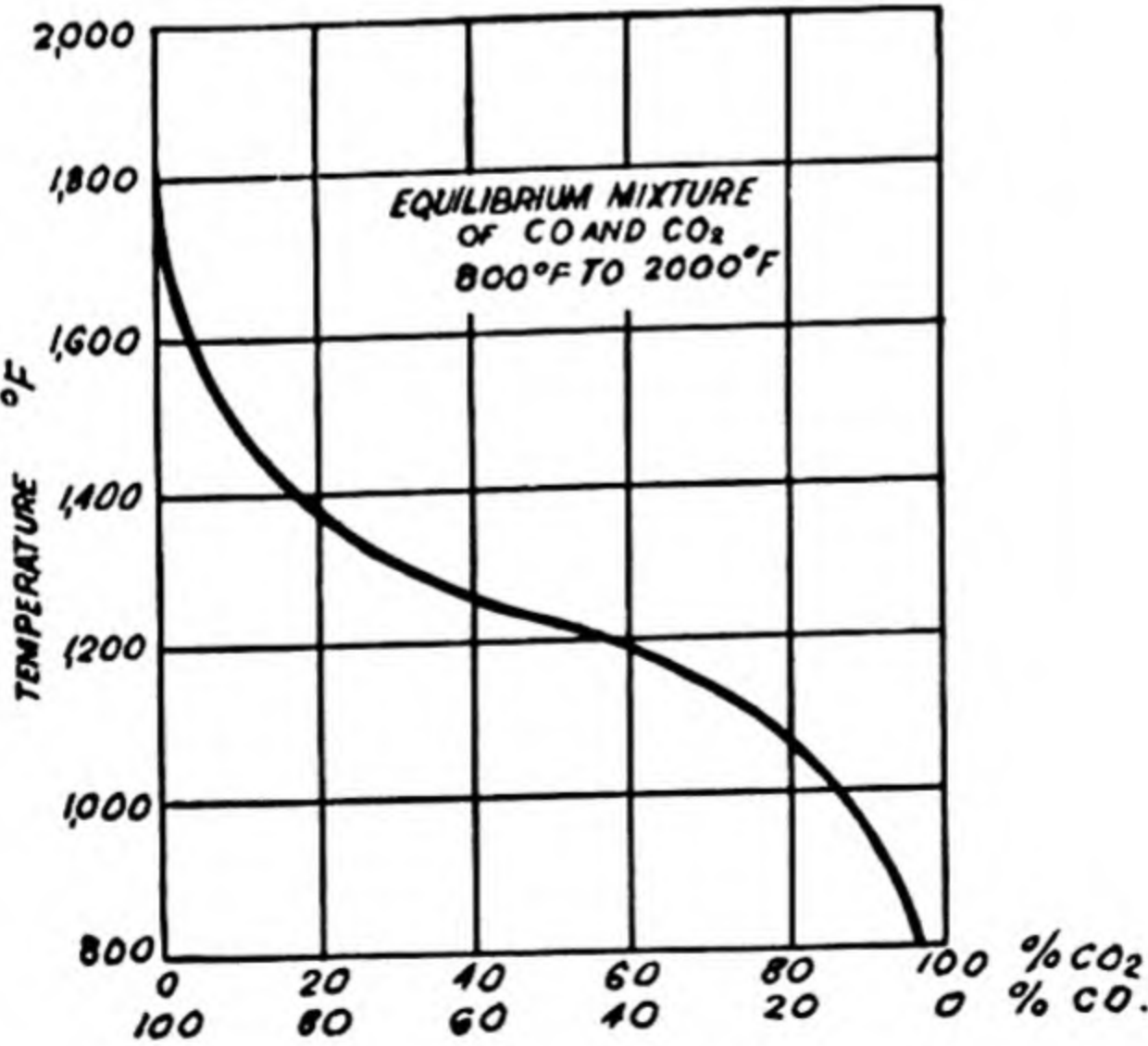


Fig. (3)3. Equilibrium Mixture of CO and CO<sub>2</sub> in the Presence of Carbon.



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where  $\Delta H$  and  $\Delta S$  refer to the heat of reaction and entropy change between reactants and products, both regarded as functions of temperature. Equations 3/21 or 3/22 can be used for the calculation of  $K$  at any desired temperature when it is known at any other temperature.

In most of the reactions which enter into furnace calculations the heat of reaction  $\Delta H$  and the entropy change  $\Delta S$  are approximately independent of temperature, except that the entropy change  $\Delta S$  alters as a temperature is reached at which the latent heat of fusion or evaporation is released. Hence the right-hand side of equation 3/22 gives approximately a series of straight lines when plotted against temperature, the slope of which changes abruptly at these points where changes of phase occur. If, now, a series of reactions are considered into all of which oxygen enters on the left-hand side and gaseous reaction products are at a pressure of 1 atmosphere, the values of  $RT \log_e p_{O_2}$  for each reaction will give a straight line when plotted on a chart with temperature as abscissa. Except where a gas is being burnt or solid or liquid solutions are formed,  $\log p_{O_2}$  will equal  $\log K$ . Thus reactions in which oxygen reacts with a solid to produce a solid will give a line through  $\Delta H$  for absolute zero of temperature sloping upwards with temperature with a slope  $\Delta S$ . They will be approximately parallel, the slope depending mainly on the latent heat of evaporation of the oxygen, since  $\Delta S$  is assumed to be zero at the absolute of temperature zero for solid  $O_2$ . When a temperature is reached at which the element becomes liquid or volatile, the slope of the line will be increased, since  $\Delta S$  is increased. When a temperature is reached at which the oxide becomes liquid or volatile, the slope will be reduced.

Carbon occupies a special position on such a diagram since both its oxides are gaseous. The combination of carbon and oxygen to make carbon monoxide thus gives a line sloping downwards, since the latent heat of evaporation of the two molecules of CO exceeds that of the  $O_2$ . Combustion with oxygen to produce  $CO_2$  gives a line which is nearly horizontal, while the combustion of CO with oxygen gives a line sloping upwards.

Now the usefulness of such a graph lies in the fact that *when at a given temperature the point on the line for one reaction lies below that on the line for another reaction, the element involved in the first reaction will theoretically be able to reduce that involved in the second at this particular temperature.* One way of picturing this is to say that the reaction occurring lower has a greater shortage of oxygen



than the one occurring higher, which it will therefore make up by reducing the oxide of the higher one. Referring therefore to Fig. (3)4, which is based on Fig. 1 of ref. 3.7, it can be seen that

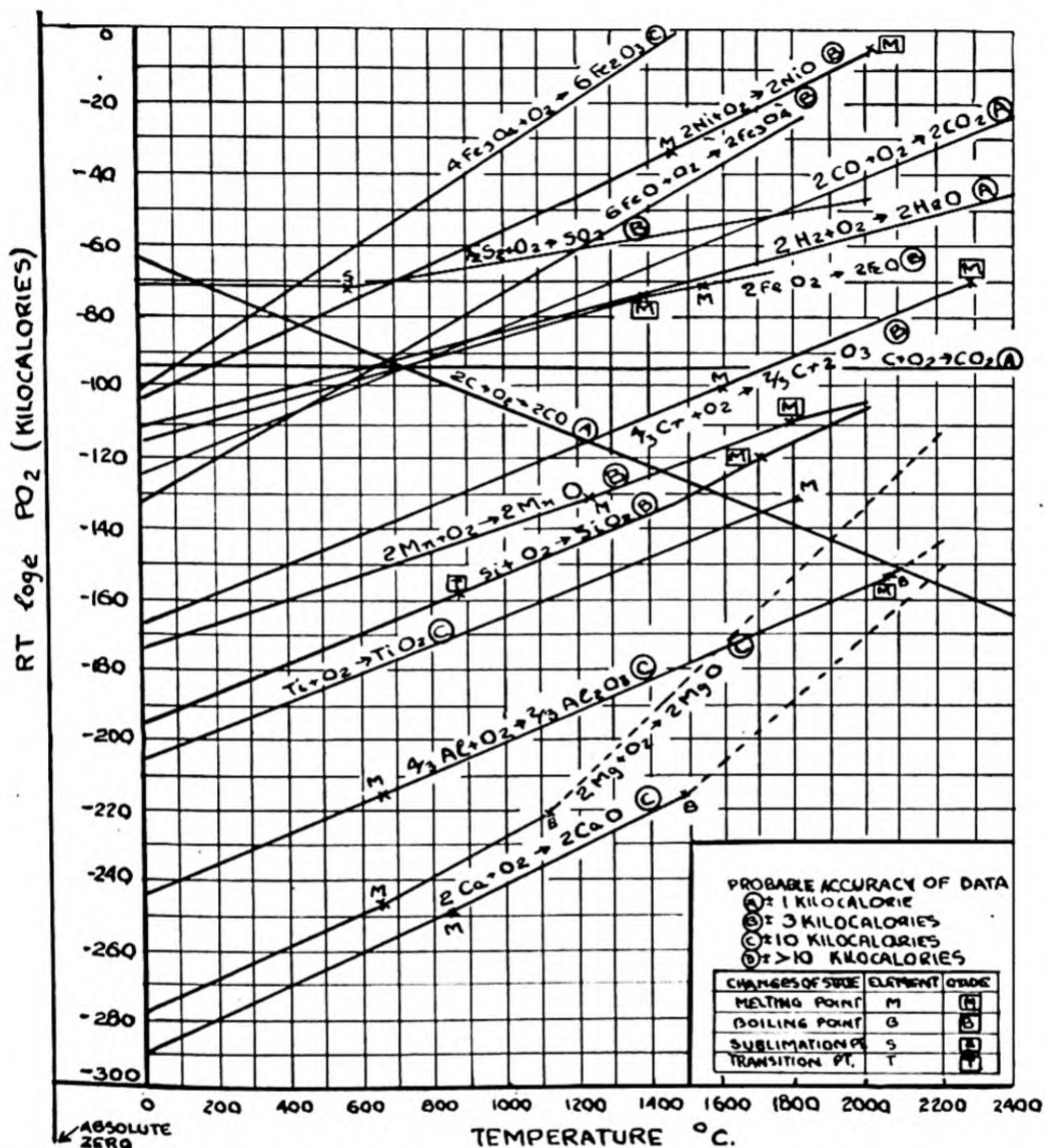


Fig. (3)4. Free-Energy/Temperature Diagrams for the Reactions of Interest in Iron and Steel Making Furnaces. (From Richardson & Jeffes. J.I.S.I. v. 160, Pt. 3, p. 261.)

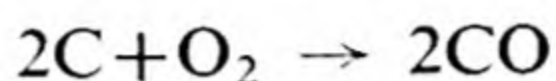
(Courtesy of the Iron and Steel Institute.)

lime is the most difficult oxide to reduce, followed at low temperatures by magnesia and alumina, while the various oxides of iron come high up. Thus at atmospheric pressure lime can only be reduced by carbon at a temperature of nearly 2200° C. Since the reaction product is CO, this reaction will be assisted by carrying it



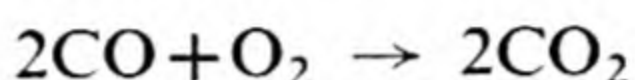
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out in a partial vacuum to remove the CO. Thus at a pressure of  $p$  atmospheres and a temperature  $T^\circ$  K. the point for the reaction



is raised by addition of  $RT \log_e p = 4.57 T \log_{10} p$ . For a partial vacuum this will be negative and the carbon oxidation point will come below the MgO oxidation point at a considerably lower temperature.

Another calculation very important in furnace operation is the proportion of CO and CO<sub>2</sub> in combustion gases which will just not oxidise a given metal. This can also be calculated from Fig. (3)2 by combining the reaction



and the oxidation reaction for the metal in question. When CO<sub>2</sub> and CO are present in a ratio of 1 : 1 the condition of equilibrium is that the line on Fig. (3)2 for this  $2CO + O_2 \rightarrow 2CO_2$  reaction crosses the oxidation line at the given temperature since the partial pressure of O<sub>2</sub> in equilibrium with the CO : CO<sub>2</sub> mixtures is equal to the partial pressure of O<sub>2</sub> in equilibrium with the metal and its oxide. When the ratio of CO : CO<sub>2</sub> is 10<sup>n</sup> the point for the reaction  $2CO + O_2 \rightarrow 2CO_2$  is at any temperature  $T$  lowered by

$$2RT \log_e 10^n = \frac{2 \cdot 4.57 nT}{1000} \text{ kcal.}$$

The reduction of less easily reduced elements will thus be assisted by keeping very low the CO<sub>2</sub> in the resulting gases.

In this way the second law of thermodynamics can be used very conveniently for calculating under what conditions elements will be oxidised in furnaces and under what conditions oxides will be reduced.

## 3.2. The Speeds of Reaction of Mixed Gases

**3.2.1. The Rates of Gaseous Reactions when Proceeding Uniformly in the Whole Volume.** The reactions of mixed gases can be studied in two ways. In the first, the mixture is introduced into a vessel at the required temperature and pressure, and the reaction is allowed to proceed simultaneously at all points throughout the volume of the gases. It is then possible to ascertain experimentally the course of the reaction by observing the change of pressure or of gas composition. This method of study is the simplest in that



gas flow and diffusion play no role and, apart from the effects due to reactions on the walls of the vessel, the conditions are uniform in space. This is termed by Lewis and von Elbe<sup>3.8</sup> "self-ignition." In the second type of system a flame or region of intense reaction is propagated through a gaseous mixture (the same authors call this "ignition by a local source"). In this case the chemical factors are complicated by the addition of physical ones such as the movement of the gas due to the expansion and diffusion of different chemical species from one point to another. Although, therefore, even the second type of reaction is simpler than those which occur in furnace systems it is necessary to study the results of experiments on the first type before an understanding of the more complicated type can be reached. In this section (3.2.1), therefore, a brief survey is given of the results obtained by studying gas reactions in which the system is homogeneous throughout. In the next three sections the more complicated systems of flame propagation will be considered.

The reaction rate is defined as the change in concentrations of the reacting substances per unit time ; thus the rate of the reaction (3/19) towards the right would be

$$\frac{1}{p} \cdot \frac{dC}{dt}, \text{ which is necessarily equal to } \frac{1}{q} \cdot \frac{dD}{dt} \quad (3/23)$$

Now gaseous reactions between molecules must occur as the result of collisions between them so that the rate will be proportional to the number of collisions and hence to the concentrations of the reactants. Hence we can write

$$\frac{1}{p} \cdot \frac{dC}{dt} = \frac{1}{q} \cdot \frac{dD}{dt} = k_1(A)^m(B)^n$$

where  $k_1$  is the reaction rate coefficient, that is, a factor depending on the fraction of collisions which are successful.  $k_1$  is accordingly a function of temperature and pressure for any given reaction but not of the concentrations. It follows from this definition that if  $k_2$  is the reaction rate towards the left, then at equilibrium (where the velocity of the reaction in either direction is equal) the equilibrium constant  $K$  is equal to the ratio  $k_2/k_1$ . Thus, just as in the case of a body in radiation equilibrium with its surroundings, considered in Chapter 4, the equilibrium is a dynamic one. The

<sup>3.8</sup> LEWIS, B., and VON ELBE, G.: "Ignition and Flame Stabilisation in Gases," *Trans. A.S.M.E.*, 70, p. 307 (1948).



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variation of reaction rates with temperature is found experimentally to be similar to that of equilibrium constants, i.e. to be of the form  $k = A \cdot e^{-q/RT}$ .<sup>\*</sup> This form of equation for the reaction rates was found experimentally by Arrhenius and it was originally thought that it could be explained in exactly the same way as the van't Hoff equation for equilibria. Thus it was considered that a reaction took place through the formation of an intermediate species of higher energy, the heat of formation of this intermediate species being  $-q$ . In this case the reaction rate would be equal to the equilibrium constant for the formation of the intermediate species and the van't Hoff equation would immediately apply.

Hinshelwood<sup>3.9</sup> and his school have developed the idea that it is more appropriate to base the explanation of the Arrhenius equation on the kinetic theory of gases rather than on the basic laws of thermodynamics. Thus Kassel<sup>3.10</sup> refers to thermodynamics as the legislative body which stipulates the changes which are permitted, while kinetics is the executive body which says what actually occurs. Hinshelwood's explanation of the Arrhenius equation is accordingly based on the idea that a certain fraction  $e^{-q/RT}$  of the gas molecules will react because they have energy greater than  $q$  by virtue of the Maxwell-Boltzmann energy distribution law applying to gas molecules at a given temperature. Eyring,<sup>3.11</sup> Polanyi and others give an explanation of Arrhenius equation somewhat similar to the original one in that they postulate a semi-stable transition state which exists owing to the presence of a trough at the summit of the curve connecting potential energy with the inter-atomic distances as the atoms approach. They attempt to go further and calculate the values of  $q$  and  $A$  in the equation

$$dm/dt = A \cdot e^{-q/RT} \quad . \quad . \quad . \quad . \quad . \quad (3/24)$$

from spectroscopic data. To do this a number of simplifying assumptions are made, including the assumptions that there are no electron transitions and that the forces are due to (1) the electro-

<sup>\*</sup> Integration of equation 3/21 with respect to  $T$  when  $H_p$  is independent of  $T$  gives

$$\log k = E - \frac{H_p}{RT} \text{ or } k = F e^{-H_p/RT}$$

where  $E, F$  are constants such that  $E = \log_e F$ .

3.9 HINSHELWOOD, C. N.: *The Kinetics of Chemical Changes* (O.U.P.).

3.10 *Kinetics of Homogeneous Gas Reactions* (Chem. Catalog Co.).

3.11 See e.g. GLASSTONE, S., LAIDLER, K. J., and EYRING, H.: *The Theory of Rate Processes* (McGraw Hill, 1941).



static attractions between the uncoupled electrons, and (2) to the resonance or exchange energies. The value of  $A$  which is a frequency factor (dimensions: 1/time) is calculated on the assumption that it is a statistical factor multiplied by the number of collisions made in unit time. As this number of collisions is proportional to  $T^{\frac{1}{2}}$  (where  $T$  is the absolute temperature) the accuracy of this assumption can be directly checked by comparing variations of the value of  $A$  deduced from experimental data. The value of  $q$  is calculated from the height of the trough in the potential energy surface, which is in turn deduced from the spectroscopic data for molecules composed of pairs of the atoms.

In certain cases good agreement of this theory with the experimental values of  $q$  are obtained, but the theory is based on rather far-reaching assumptions and the explanation is by no means fully accepted (see Hinshelwood, pp. 238 and 239).<sup>3.9</sup> However, it can be said that wave mechanics is being used to a useful extent to explain the mechanism of combustion.

In certain other cases, however, the Arrhenius equation and the corresponding equation for the effect of pressure do not give a correct representation of the variation of the rates of reaction of homogeneous gas mixtures with the temperature and pressure. Thus, for example, it has been found that mixture of hydrogen and oxygen will explode below a certain critical pressure and above another critical pressure. Explosions caused by an increase of pressure could be explained by the hypothesis that explosion results from the increase of reaction rate with temperature, coupled with self-heating of the gas by the reaction (i.e. an energy-chain explanation), but the case of an explosion resulting from a decrease in pressure cannot be explained on this basis. It has therefore been necessary to postulate that homogeneous gas reactions do not in general take place in the manner indicated by the simple equations, but depend on the formation of intermediate particles which carry abnormal amounts of either chemical or thermal energy. Under certain conditions the number of active intermediate products formed in the series of reactions exceed the number removed by the series, and in this case the reaction is self-accelerating and an explosion can take place. The intermediate particles are referred to as chain-carriers; reactions which eliminate them are called chain-breaking reactions, and those which increase the number of them chain-branching reactions. It is then possible to explain the existence of a pressure below which explosion occurs by the fact



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that increase of pressure results in an increase of the importance of chain-terminating reactions. The walls of the vessel and objects inserted into the vessel can also serve in certain of the reactions for chain breaking. Chain breaking by means of vapours or gases introduced into the mixture in small quantities becomes also relatively easy to understand.\*

With this type of explanation it is possible in many cases to work out reaction schemes which give a good account of the observations both with regard to the existence of explosion regions on the pressure temperature curve and as regards the speed of the reactions. In particular the combustion of hydrogen and carbon monoxide and that of a number of hydrocarbons has been studied.

In the combustion of hydrogen with oxygen the chain-carriers are mainly  $\text{HO}_2$  and  $\text{H}$  and the reaction can under certain conditions be made explosive by means of small quantities of  $\text{NO}_2$  and is always inhibited by iodine vapour. In the case of carbon monoxide the reaction is very sensitive to the presence of a small quantity of water vapour, which increases greatly the rate of reaction although it does not affect the position of the explosion limits. As with hydrogen,  $\text{NO}_2$  changes the reaction limits in the direction of increased explosion regions, while iodine in many cases inhibits reaction. The reactions of hydrocarbons are extremely complicated and probably involve the formation of formaldehyde and  $\text{OH}$  radicals, and possibly other radicals as well, e.g.  $\text{CH}_3$ ,  $\text{CH}_2$ , etc.†

At higher temperatures, i.e. over  $1000^\circ \text{C}$ ., all reactions are very much faster. When hydrogen and carbon monoxide are allowed to burn simultaneously with excess of oxygen, hydrogen burns about three times as fast as  $\text{CO}$ . The combustion of hydrocarbons remains, however, a complex process involving the formation of formaldehyde which breaks down thermally to form  $\text{CO}$  and  $\text{H}_2$  or burns immediately to  $\text{CO}$  and water or to  $\text{CO}_2$  and water according to the amount of oxygen present. In these cases no soot or luminosity is formed. Where there is a deficiency of oxygen‡ there is a race

\* This is an illustration of the close connection between researches into chemical kinetics and the internal combustion engine, where the introduction of lead tetraethyl prevents detonation in the cylinder (knock).

† A detailed discussion of these reactions is given in LEWIS and VON ELBE, *Combustion Flames and Explosion of Gases*, and the use of spectroscopic methods to determine the stages of combustion and the radicals formed is given by GAYDON in his recently revised book, *Spectroscopy and Combustion Theory* (Chapman and Hall).

‡ In diffusion flames there is necessarily a deficiency of oxygen where combustion starts.



between cracking of the hydrocarbons into carbon (soot) and hydrogen, and the combustion process via formaldehyde. Norrish and Patnaik<sup>3.12</sup> have recently shown that the combustion of methane is accelerated by ultra-violet light of the wavelength corresponding to the absorption spectrum line of formaldehyde.

**3.2.2. Flame Propagation Through a Gaseous Mixture.** In the case of the more complex type of flame of the two mentioned at the beginning of Section 3.2.1, theory has not been so successful in explaining the experimental observations because the mechanical processes are, as suggested above, complicated by physical ones. Two types of experimental arrangements have been used to study the speeds of reaction under conditions of flame propagation. In the one, the gaseous mixture is enclosed in a rigid spherical vessel or in a soap bubble and is ignited from a central point, the rise of pressure in the sphere being measured in the former case and the actual expansion of the visible flame sphere being observed in the latter. The results of experiments of this type are discussed in this section (3.2.2). In the second case the steady-state condition where a gas flows through a system and burns as it passes through a certain region has been studied. This corresponds to the industrial cases of the Bunsen burner and of surface combustion which are dealt with in Sections 3.2.3 and 3.2.4.

The first observation of importance in the stationary gas experiment is that there are certain limits of mixture ratio beyond which flames are not propagated at all. These are known as *limits of inflammability* and naturally extend on either side of the stoichiometric mixture\* to the point where the mixture is too rich or too weak to ignite. The limits depend somewhat on the method of ignition and the shape of the vessel and whether the gas is static or flowing.† The existence of these limits is due to the fact that ignition proceeds by an accumulation of thermal energy and hence

\* That mixture which can be burnt completely to give no excess of combustible or oxygen.

† It has recently been shown<sup>3.13</sup> that the introduction of very fine metal particles into hydrogen and air mixtures widens the inflammability limits and lowers the ignition temperatures. Since such fine particles are always liable to be present in industrial flames they provide a further reason for regarding the laboratory measurements of ignition ranges as indicative of general trends only.

<sup>3.12</sup> NORRISH, R. G. W., and PATNAIK, D.: "Effect of Light on the Combustion of Hydrocarbons," *Nature*, p. 883 (4 June 1949).

<sup>3.13</sup> BURGOYNE, J. H., and THOMAS, N.: "Effects of Very Fine Solid Particles on Flame Propagation," *Nature*, 163, p. 765 (14 May 1949).



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their position for different mixtures will be determined by the heat of the reaction, and by the thermal conductivities, rates of diffusion and specific heats of the components. Starting from the stoichiometric mixture and proceeding towards leaner and richer mixtures, it is clear that points will be reached where the heat release in a small volume element will be insufficient to propagate flame by thermal or molecular diffusion. This is a simple physical explanation for the existence of the limits.

The limits of inflammability for hydrogen and air at room temperature and atmospheric pressure are 4 and 74% of hydrogen in the mixture, those of carbon monoxide and air are 12.5 and 74%

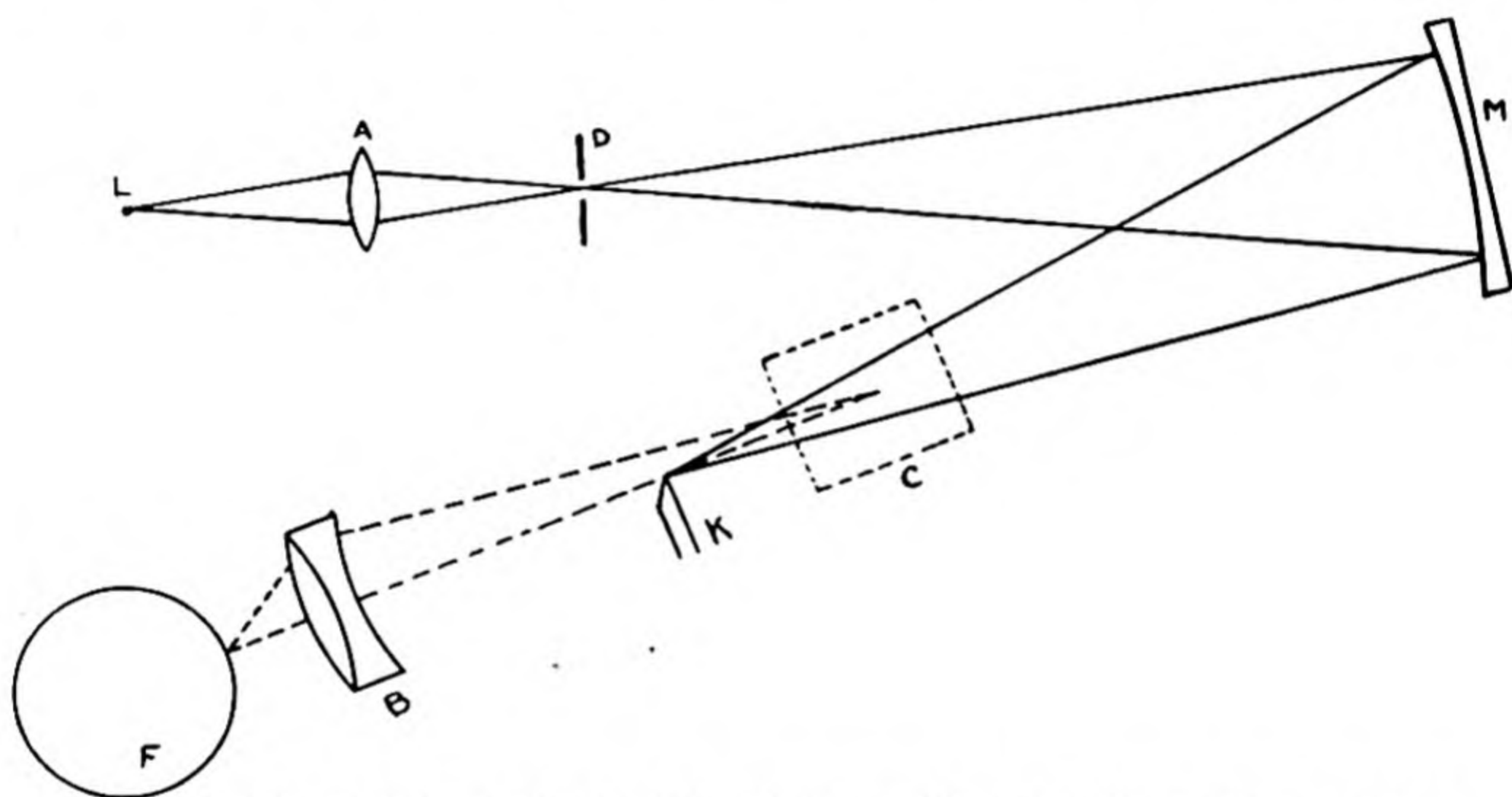


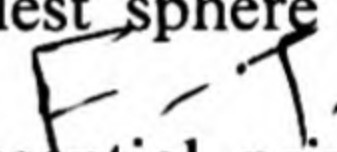
Fig. (3)5. The Schlieren Method for Photographing Density Differences in a Gas Stream or Flame.

and those of methane and air 5 and 15%. In most cases the limits of inflammability correspond, as one would expect, to temperatures produced by combustion at which the rate of combustion would be very fast. The exception is the lower limit of inflammability of a hydrogen-air mixture, which is one the combustion of which would give only 350° C., a temperature at which the combustion would proceed very slowly. This exception has been explained as being due to the great differences between the rates of diffusion of hydrogen and air gases ( $N_2$  and  $O_2$ ), which results in a local increase in concentration of  $H_2$  when combustion takes place.

In studying the propagation of flames in bubbles and in the steady-state burner, various techniques have been developed, one of which is of considerable potential value in the study of furnaces.



This is the Schlieren method. Light from a carbon arc is condensed by the lens  $A$  (see Fig. (3)5) on to a diaphragm  $D$ , whose image, produced by the concave mirror  $M$  is just intercepted by a knife-edge  $K$ . The combustion zone  $C$  being studied lies between  $M$  and  $K$ , and a lens  $B$  focuses the region  $C$  on to a photographic film at  $F$ , which is usually a high-speed recorder. Changes in refractive index at  $C$  cause the light to be deviated, so that this light passes the knife-edge on to the film. In this way beautiful figures of convection flow can be obtained. The method should be more widely applied to all types of combustion phenomena.

Lewis and von Elbe<sup>3.8</sup> determined the minimum energy of a spark required to ignite mixtures of methane and other hydrocarbons with oxygen and nitrogen. They found this energy to pass through a minimum when the mixture ratio was varied from weak to rich and that dilution by an inert gas has a very large effect upon it, the minimum spark energy for a stoichiometric mixture with oxygen being 1/100 that for a similar mixture with air. It was shown by calculation that the measured energies corresponded to the energy which would be required to heat unit volume of the reactants to the point where the energy of combustion can further heat them multiplied by the volume of the smallest sphere from which combustion is capable of self-propagation. 

**3.2.3. Combustion at a Burner Mouth.** The essential principle of the Bunsen burner is that a gas-air mixture flows up a tube at a speed greater than the velocity of flame propagation in this particular mixture. As the velocity of the gas at the rim of the tube is zero, the flame can attach itself at this point and the gas burns across a cone such that the component of the gas velocity normal to its surface equals the burning velocity. The burning velocity  $V$  can be determined by measuring the angle  $\alpha$  of the cone at a point at a distance 0.707 of the radius of the burner tube from the axis, since at this point the flow velocity rate  $u$  equals the average velocity over the whole cross section ( $\sin \alpha = V/u$ ). Outside the cone the reaction proceeds to completion by means of secondary air which diffuses in from the outside. Fig. (3)6 shows the burning speed for various mixtures of combustible gases and air at atmospheric pressure.

Lewis and von Elbe<sup>3.8</sup> have discussed in greater detail the mechanism by which the flame attaches itself to the rim of the burner. The burning velocity of the premixed gas is constant across the centre of the gas stream, but falls sharply to zero at a



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certain distance from the cold tube, called the quenching distance. The gas flow velocity, on the other hand, falls off to zero linearly with distance from the wall in the region of the wall. If this curve cuts the burning velocity curve, the burning velocity exceeds the gas velocity at a certain distance from the wall and the flame flashes back. If it is tangential to the curve, which can happen at only one gas flow rate, then the flame is exactly stable at the burner mouth. This is the lower limit of stability. If the flow rate exceeds this value the flame is blown away from the burner mouth to a point where the burning velocity curve is tangential to the flow curve, because the former remains high further out as the flow cross-section is taken further away from the tube end. The burning

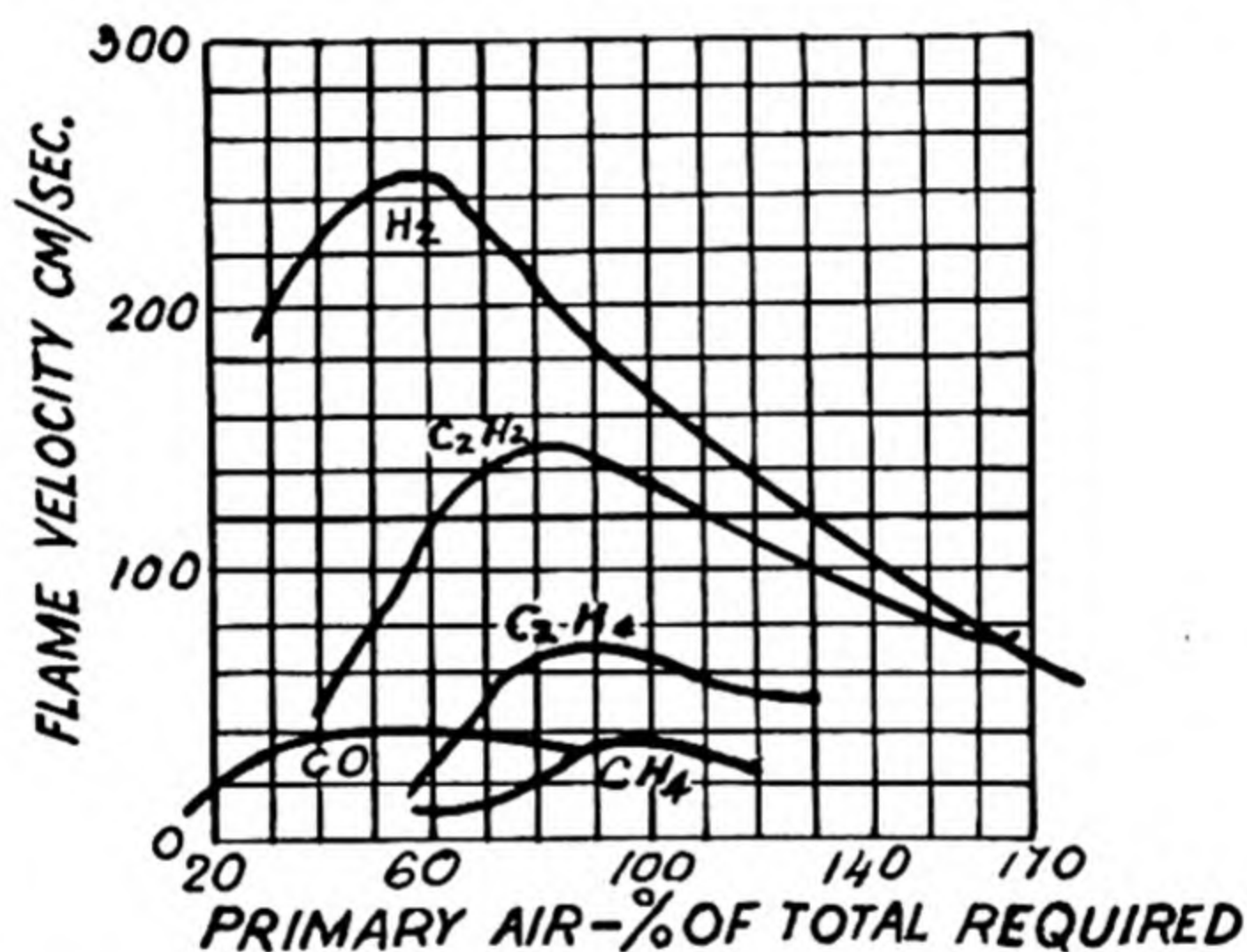


Fig. (3)6. Burning Speeds of Combustible Gases mixed with air.

velocity does not, however, retain its high value across a width as high as the tube diameter at all distances from the burner mouth, because of interdiffusion of air, and hence there is a certain upper limit of stability above which the flow velocity exceeds the burning velocity at all points and the flame is blown away. This theory has been confirmed experimentally by showing that the curve of critical flashback velocity gradient at the boundary against mixture composition is independent of the tube diameter over a range of tubes between very small ones where the quenching distance is comparable with the radius (0.3 cm. diam.) and large ones where the flame flashes back irregularly (1.5 cm. diam.).

The mechanism of the Bunsen flame is thus fairly well understood, but the actual value of the burning velocity is very difficult to



calculate theoretically from the reaction rates, as it depends on diffusion also ; hence although a number of workers have been able to measure burning velocities and obtain quite good agreement, these results remain empirical. Lewis and von Elbe<sup>3.1</sup> have, however, been able to give a theoretical treatment of the simple case of the decomposition of ozone. Earlier attempts to give a theory based on the existence of an ignition temperature  $T_i$  and of conduction of heat from the burnt to the unburnt gas are obviously open to the criticism that modern experiments discussed in Section 3.2.1 show that there is no such ignition temperature. These earlier attempts lead to the equation

$$\text{Flame speed} = \frac{k}{\rho_u \sigma_p} \cdot \frac{T_b - T_i}{T_i - T_u} \cdot \frac{1}{X_b} \text{ cm./sec.} \quad (3/25)$$

where  $k$  = thermal conductivity of gas cal./cm. sec.° C.

$\rho_u$  = density of unburnt gas.

$\sigma_p$  = average specific heat at constant pressure (i.e. average before and after).

$T_u, T_b$  = temperature of unburnt and burnt gases.

$X_b$  = thickness of reaction zone.

This equation predicts correctly the existence of limits of inflammability (since for very lean or rich mixtures  $T_b$  will fall as low as  $T_i$ ), but does not give satisfactory agreement ; for example, on the effect of changing the diluent gas so as to change the heat conductivity of the mixture.

In their calculation Lewis and von Elbe allowed for the existence of active particles which diffuse against the gas flow and thus assist the heat flow in propagating the reaction. They found it necessary to make the simplifying assumption that the sum of the thermal and the chemical energy per unit mass at all points is constant. With this assumption they were able to calculate a burning velocity which gave agreement as to order of magnitude with the observed values and the variation with pressure and temperature. Their calculation showed that the thickness of the flame front is only of the order of  $10^{-3}$  cm. and that the chain carriers (oxygen atoms) rise to a quite high concentration in this narrow region.

Tanford<sup>3.14</sup> has discussed the diffusion of heat and matter in the neighbourhood of the flame front for mixtures of oxygen and nitrogen with moist CO or hydrogen in the attempt to calculate the

<sup>3.14</sup> TANFORD, C. : "Theory of Burning Velocity," *Jl. Chem. Phys.*, **15**, pp. 431 and 861 (1947).



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burning velocity. He showed that the temperature falls rapidly with distance from the flame front and hence that the effect of molecular diffusion of hydrogen atoms reaches much further from the flame front than does that of thermal diffusion, so that the former is the more active process in propagating combustion. A similar conclusion has been reached by Gaydon and Wolfhard<sup>3.15</sup> from their experimental results that in stationary premixed hydrocarbon flames the thickness of the reaction zone varies inversely with the pressure, but the flame speed is independent of pressure. They calculated that the time of passage through the preheating zone is very much less than the delay or induction period for the temperature (700°–800° C.) at which marked exothermic reaction begins. All these results could be accurately explained if the flame is assumed to be propagated by diffusion of free atoms or radicals, probably free hydrogen atoms being the active species. The final formula for the flame velocity  $v$  is

$$v^2 = \frac{D \times \bar{u}}{2n_c} \cdot \frac{T_u}{T_r}$$

where  $\bar{u}$  is the mean reaction velocity  $= n_c v / X_b$  ( $X_b$  is thickness of the reaction zone) ;

$D$  is the diffusion coefficient for the active species diffusing into the unburnt gas ;

$n_c$  is the number of combustible molecules/cm.<sup>3</sup> present initially ;

$T_u$  is the initial temperature of the mixture ;

$T_r$  is the mean temperature in the reaction zone.

**3.2.4. Surface Combustion.** In the case of surface combustion the gas-air mixture is projected at such a high velocity through a nozzle that it does not burn at all in contact with the nozzle, but burns where it comes into contact with a refractory surface which forms a region of lower velocity where the reaction can start. In this case the refractory is maintained at a very high temperature and this combustion is effectively instantaneous. The flame speed which must be exceeded by the gas leaving the burner mouth has to be very much higher than in the case of the Bunsen type of burner or combustion will start from the rim of the nozzle.

<sup>3.15</sup> GAYDON, A. G., and WOLFARD, H. G. : "The Influence of Diffusion on Flame Propagation," *Proc. Roy. Soc. A.*, **196**, p. 105 (1949).



### 3.3. The Speed of Combustion of Diffusion Flames

**3.3.1. Examples of Diffusion Flames.** By diffusion flame is meant a flame in which combustion follows from the gradual mixing of air and the combustible gas after these have been introduced separately into the combustion region. Clearly, premixing of the air and gas is not possible where it is desired to preheat one or both of these before combustion, since such preheat would cause premature ignition, and one is therefore obliged to use diffusion flames in such cases. Generally speaking, diffusion flames are considerably longer than premixed flames because of the difficulty of rapidly obtaining complete mixing in the furnace chamber itself. The simplest example of a diffusion flame is the luminous flame obtained when the air inlet hole of a Bunsen burner is blocked, an example which illustrates the fact that luminosity is characteristic of diffusion flames where hydrocarbons are among the combustible materials. Another very important example is the flame of producer gas in an open-hearth furnace, where the gas, preheated to  $1000^{\circ}$ – $1400^{\circ}$  C., enters through a port about  $2\text{ ft.}^2$  with a velocity of  $100$ – $200\text{ ft./sec.}$  while the air similarly preheated enters through a much larger horseshoe-shaped port above and around the gas port.

It is shown in 3.3.2 and 3.3.3 below that the factor governing the speed of combustion in diffusion flames is the mixing of the air and fuel which is necessarily slow compared with the combustion speed of the fully mixed hot gases. Hence the different types of diffusion flame burners represent simply different methods of controlling the mixing of the air and gas. The most common type is that in which the gas fuel is injected in a high velocity jet which entrains the surrounding low velocity combustion air as a consequence of its own momentum (see Section 5.6.2). All open-hearth furnace port systems work essentially on this principle. Another common method, studied essentially by Rummel, is that in which the air and gas jets issue in two streams which impinge on one another. The third category is those in which the air or gas streams are concentric, like the first, but one or both are given a swirling or rotary motion in order either to accelerate mixing mainly by increasing the velocity gradient, or to control the flame direction. This is most usually done by fixed passages or blades, as for example in the Teisen<sup>3.16</sup> tangential burner where the air enters tangentially the

<sup>3.16</sup> TEISEN, TH.: *Trans. Soc. Glass Tech.*, **11**, p. 265 (1927).



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vertical brick flue carrying the gas, but in some, such as the Fan-mix,<sup>3.17</sup> the air is actually made to rotate a moving arm which carries the gas inlet nozzles.

Because of the difficulty of attaining complete combustion in a diffusion flame, it is generally necessary to operate such flames with appreciably more excess air than would be necessary with premixed flames, otherwise a mixture of oxygen and unburnt combustible will be found in the exit gases. The main aim of research on such flames from the point of view of furnace design is therefore to find ways of controlling the mixing and thus shortening the flame. In small-scale flames, conditions may be appropriate for streamline flow, and in this case the combustion takes place as a result of direct molecular diffusion. Such flames have been studied in detail by Burke and Schumann, whose treatment is discussed in Section 3.3.2. In larger furnaces, on the other hand, the mixing is mainly by means of "eddy" diffusion, which is the transfer by turbulence of whole pockets of air and gas across the boundary surface. This subject has been studied particularly by Rummel and by Heiligenstadt in Germany and is discussed in Section 3.3.3, while Section 3.3.4 deals with practical methods of investigating mixing conditions.

**3.3.2. Streamline Diffusion Flames.** Burke and Schumann<sup>3.18</sup> carried out a complete mathematical investigation of the case of a combustible gas flowing upwards out of a tube into a larger one through which air or oxygen is flowing at the same velocity. They calculated both the case of concentric circular tubes and of flat flames, making use of certain simplifying assumptions. They found that the zone of reaction was very thin, so that it could be treated as a geometrical surface. For the purpose of calculation, this zone was taken as the locus of the points where the rate of diffusion of combustible gas outward and that of oxygen inward have the ratio required for complete combustion. They were thus able to calculate the shape of the flame front, and found not only that, as would be expected, it is a curved surface moving to the central axis in the case of an over-ventilated flame and to the outer walls in the case of an under-ventilated flame, but also these calculated surfaces were found to agree closely with the experimental shapes. They also

3.17 GRISWOLD, J.: *Fuels, Combustion and Furnaces*, p. 271 (McGraw Hill, 1946).

3.18 BURKE, S. P., and SCHUMANN, T. E. W.: "Diffusion Flames," *Ind. Eng. Chem.*, **20**, p. 998 (1928).



concluded that the height of a cylindrical flame should not be affected by the size of the tubes if the volume flow of air and gas is kept constant. This was found experimentally to be true. Qualitative agreement was obtained with the effect of changing the combustible gas, e.g. a carbon monoxide flame is 2.5 times as high as a hydrogen flame, while they calculate it should be four times as high. Accurate agreement was obtained with their prediction that the vertical dimensions of a flame should be directly proportional to the velocity of flow. They also predicted that the effect on the height of the flame of preheating the gases should be zero and it was in fact found to be very slight. These results are sufficiently good to verify their general conclusion that the rate-determining factor in this type of flame is the physical process of diffusion and that the mechanism and rate of reaction are of secondary importance, a conclusion which, as described in the next section, also holds for two-phase combustion reactions (e.g. combustion of pulverised coal) above a certain temperature.

**3.3.3. Turbulent Diffusion Flames.** The effect of various factors on the combustion of gas and air in furnaces has been studied in detail by Rummel<sup>3.19</sup> and by Heiligenstadt.<sup>3.20</sup> Rummel's work is valuable chiefly in that it provides a qualitative account of the effects of various factors in controlling the combustion volume, while Heiligenstadt gives numerical formulæ which are useful for calculating combustion chambers with conventional burners. Thus Rummel's work is of value to the designer who wishes to *change* the combustion volume by altering the burner, while Heiligenstadt's formula is valuable in designing with a standard burner. Rummel used both a full-scale combustion furnace and also small-scale air models, but could not, as in the case of Burke and Schumann, give a theoretical treatment since the laws of diffusion by eddies are much more complicated than those for molecular diffusion. The most important conclusion from Rummel's first part is the same as that obtained by Burke and Schumann, namely, that the speed of combustion is almost entirely governed by the speed of mixing. This is shown by the fact that the effect of temperature on combustion between 900° C. and 1300° C. is not noticeable and that it is likely that there is no significant effect

<sup>3.19</sup> RUMMEL, K.: *Mitt. Nr. 242 der Wärmestelle Archiv. f.d. Eisenhüttenw.*, 10, p. 505; 11, pp. 75, 113 (1936).

<sup>3.20</sup> HEILIGENSTADT, W.: *Wärmetechnische Rechnungen für Bau und Betrieb von Öfen* (Springer, Berlin).



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between temperatures as widely separated as  $600^{\circ}\text{C}$ . and  $1800^{\circ}\text{C}$ . (Below  $600^{\circ}\text{C}$ . the chemical reaction rate would play a role, while the higher temperatures would cause a greater spreading out of the combustion gases and hence the necessity for a greater volume for mixing.) Rummel proved that the actual process takes place mainly by turbulence rather than by molecular diffusion, by calculating that with gas and air layers of 10 cm. thickness, 50 seconds would be required to obtain 98% combustion if diffusion alone came in, whereas experiment shows that it actually occurs in a time which is only a small fraction of this.

The following general conclusions as to the effect of various control variables on the flame length were reached by Rummel for air and gas jets emerging from ports with rectangular cross-sections, the longer sides of each being equal in length and situated side by side.

- (i) Introduction of the two gas streams side by side with *very different velocities* reduces the combustion volume greatly as compared with that necessary when they have equal velocities, and so does increasing their velocities in the ports.
- (ii) Buoyancy effects due to temperature differences are small, but buoyancy of a gas stream that is lighter than air at the same temperature (usually due to the presence of hydrogen) does accelerate mixing when the gas is introduced below the air.
- (iii) A thick tongue of brick between the air and gas streams increases the flame length very much, whereas introduction of the gases in rapidly converging, flat streams is the most powerful means of reducing it. For example, the mixing space was only one-sixth as large when the jets were at  $90^{\circ}$  to one another as when they were parallel.
- (iv) Combustion at the end of a flame can be accelerated by impact on a wall or passage round a narrow bend.
- (v) Excess air accelerates combustion of the gas during the whole process and not only at the end of the flame.
- (vi) The general shape of the flames was found to be rather similar to that of Burke and Schumann, namely, a surface spreading from the mouth of the burner bent to one side or the other according to whether there is excess air or excess gas.



Recently Hawthorne, Weddell and Hottel<sup>3.21</sup> have studied the flame length of turbulent gas jets entraining their combustion air from stagnant surroundings. They have shown that here again the mixing speed governs the flame length because the mixing length calculated from jet formulæ agrees closely with the measured flame length. For turbulent free flames unaffected by buoyancy they found the following quantitative formula giving the flame length  $L$

$$L/D = \frac{5.3}{C_r} \sqrt{\frac{T_F}{\alpha_T T_N} \left[ C_r + (1 - C_r) \frac{M_S}{M_N} \right]}$$

where  $L$  = visible flame length, ft. or cm.

$D$  = nozzle diameter, ft. or cm.

$T_F$  = adiabatic flame temperature, °K.

$T_N$  = absolute temperature of nozzle fluid.

$M_S, M_N$  = molecular weights of surrounding and nozzle fluids, respectively.

$C_r$  = mol. fraction of nozzle fluid in the unreacted stoichiometric mixture.

$\alpha_r$  = mols of reactants/mols products, for the stoichiometric mixture.

Heiligenstadt bases his formulæ on the assumption that the volumetric combustion rate is proportional to the distance from complete combustion; he expresses the extent of combustion in terms of the mean sensible heat content of the gas per unit volume at time  $t$  seconds after it enters the combustion chamber and obtains the formula

$$i = i_0(1 - e^{-kt}) \text{ C.H.U./} n \text{ ft.}^3 \quad (3/26)$$

where  $i_0$  is the sensible heat content of 1  $n \text{ ft.}^3$  of combustion gas due to complete combustion.

$k$  is a combustion rate coefficient ( $\text{sec.}^{-1}$ ) lying between 1.5 and 10 according to the burning conditions.  $k$  must be determined experimentally for the type of burner being used, but the following values may be taken as a guide.

- (1) Brick burners—nearly parallel streams, 1.5–2.
- (2) Brick burners—streams running together, 2.5.
- (3) Metal burners—low velocity, 3–4.
- (4) High-velocity metal burners, 4–6.
- (5) Premix burners, 8–10.

<sup>3.21</sup> HAWTHORNE, W. R., WEDDELL, D. S., and HOTTEL, H. C.: "Mixing and Combustion in Turbulent Gas Jets." *Third Symposium on Combustion, Flame and Explosion Phenomena*, p. 266. (Williams & Wilkins Co., Baltimore, 1949.)



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Although this formula has not a sound theoretical basis it is claimed by Heiligenstadt to be useful because furnaces designed by means of it do work satisfactorily. Since  $k$  must be determined experimentally the only special feature of the formula is the type of time dependence. This will always be sufficiently accurate over the range of times concerned, although an accurate experimental investigation over a wider range would be of great interest.

The combustion temperature can then be calculated from this formula using the I.T. Diagram (Chapter 2, Fig. (2)1). Allowance must be made for any additional heat supplied by preheat and any given up to the charge or the walls during the time  $t$ , by expressing both these in terms of C.H.U./ $n$  ft.<sup>3</sup> of *combustion gas*. The time  $t$  taken for the gas to pass from the inlet to a specified point in the combustion chamber is calculated from the volume of heated gas per unit time and the *filled* cross-sectional area of the chamber at each intermediate point.

For furnaces with separate combustion chambers where combustion has to proceed to a specified point before entering the heating chamber, Heiligenstadt develops from the previous formula the more easily applied relation

Necessary combustion volume =  $\frac{Q}{k} \cdot \alpha$  ft.<sup>3</sup> . (3/27)

where  $Q$  = heat supply in 10<sup>6</sup>C.H.U./hr.  
 $\alpha$  is a coefficient which is given in terms of the desired fractional heat release  $\frac{i-i'}{i_o}$  and the preheat ratio  $\frac{i'}{i_o}$  ( $i$  = sensible heat supplied by preheat/ $n$  ft.<sup>3</sup> of combustion gas) by Table 3.2.

TABLE 3.2  
COMBUSTION VOLUME COEFFICIENT  $\alpha$  (HEILIGENSTADT)

$\frac{i-i'}{i_o} \backslash \frac{i'}{i_o}$	0	0.2	0.4	0.6	0.8	1.0
0.50	13	19	26	30	37	42
0.60	19	27	35	43	51	59
0.70	30	40	50	61	70	82
0.80	47	59	79	86	101	115
0.85	58	79	91	107	123	139
0.90	77	96	115	136	155	176

An attempt to calculate the combustion rate in turbulent diffusion



flames has recently been made by Wohlenberg.<sup>3.22</sup> Clearly, the assumption made by Rummel that combustion proceeds instantaneously once mixing is achieved cannot be exactly true, although when the preheat temperatures are very high, as in the open-hearth, and mixing is slow, it gives a very nearly correct picture. Wohlenberg, however, does not make this assumption, but assumes that mixing proceeds by molecular diffusion across an interface between fuel-rich and oxygen-rich zones, the extension of this interface by folding and pocketing due to turbulence being very large. The reaction rate per unit volume is thus the triple product of this interface per unit volume by the number of collisions per unit area of it by the fraction of these collisions which are effective. The first of these three terms thus allows for turbulence, the second for the ultimate molecular diffusion across the interface and the third for the probability of chemical reaction on collision. Rummel's work amounted to assuming that this third term is unity and combining the other two into a single mixing index which he determined by a model. Wohlenberg shows that when mixing is instantaneous, the maximum possible combustion rate for natural gas burnt with 1-2 times theoretical air to within 99% of completion is about 1.2 million C.H.U./hr.-ft.<sup>3</sup> of combustion space, so that when, as in the open-hearth, the actual rates are much lower, it is clear that the mixing process is the chief bottleneck.

It would seem that for such cases as the open-hearth furnace and the glass tank furnace, where the air is highly preheated, Rummel's treatment is quite satisfactory and the course of diffusion combustion can be predicted from the course of mixing studied by means of models, as described in Section 5.7.4. No fully satisfactory theoretical formulæ are available for such calculations and proper prediction of combustion rates and the effect upon them of such alterations in design must necessarily be made by means of models. For combustion rates with cold air a model can only give the mixing rate and it is necessary to modify its predictions by some calculations such as those of Wohlenberg before the actual combustion rate can be derived.

**3.3.4. Practical Methods of Studying the Combustion Conditions in Diffusion Flames.** When it is found that a furnace must be operated with more than 20% excess air before the presence of

<sup>3.22</sup> WOHLBERG, W. J.: "The Influence of Reaction Interface Extension in the Combustion of Gaseous Fuel Constituents," *Trans. A.S.M.E.*, p. 143 (April 1948).



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combustion in the outgoing ports can be avoided, the cause is almost certainly inadequate mixing of the air and gas streams due to neglect of the mixing laws enunciated in the previous section. Two methods are available to show such bad mixing experimentally and to test the effect of design modifications upon the mixing. One of these is the dilution method discussed in Section 5.7.3, where a small quantity of traceable material is introduced either into the air or the gas stream and the distribution of this material across the exit port measured. For combustion studies, however, it is often sufficient to insert water-cooled gas-sampling probes in the exit ports and take samples of the combustion gases at various points across these ports. If the gas analysis across the exit port indicates marked stratification, then direct proof has been obtained of the existence of bad mixing. As a result of a number of experiments it has been found that a satisfactory experimental arrangement for

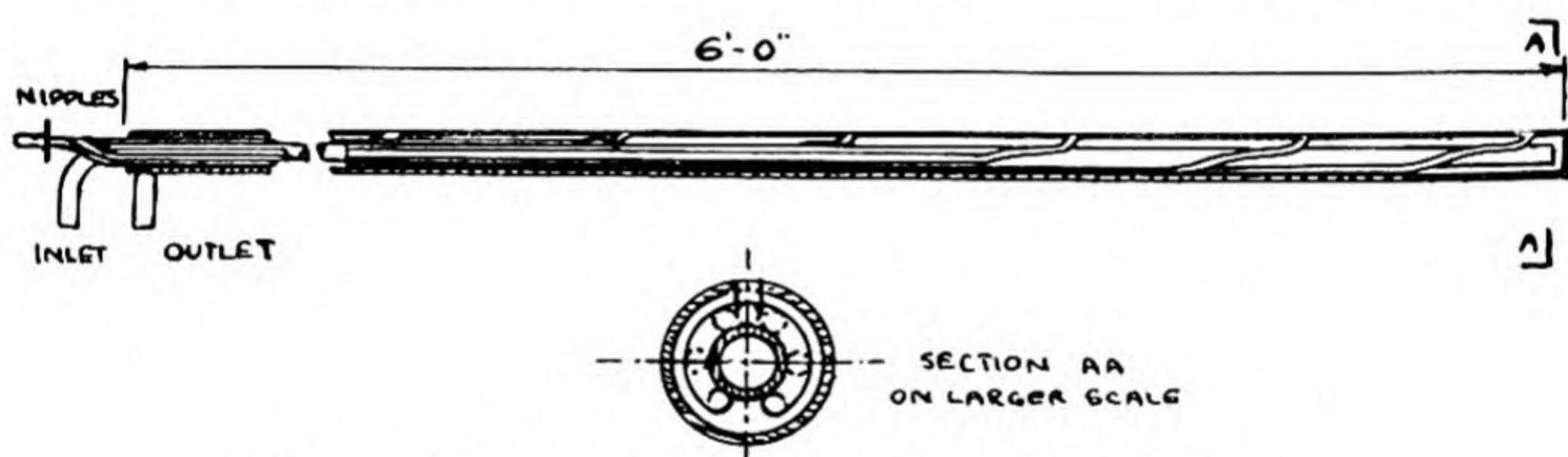


Fig. (3)7. Multi-Point Water-cooled Gas Sampling Probe.

this purpose is a multi-point water-cooled probe (see Fig. (3)7) from which samples can be drawn off simultaneously over a period long enough to obtain average results.

It will frequently arise when such a technique is used that it is desired to compare the mixing in a "good" furnace operating with less excess air with that in a "bad" furnace which requires more excess air. This means that the average  $\text{CO}_2$  across the whole height is not the same. In this case the following method can be used to give a useful criterion of mixing. Suppose for the sake of argument that the outgoing gases are divided into two fractions by means of a horizontal plane halfway up the port at the sampling point. Then the *assumption* is made that these two fractions represent equal N.T.P. volumes of gas ( $\nu$  ft.<sup>3</sup>/min.). Let  $p_1, q_1$  denote the N.T.P. volume of air and gas respectively that react to give  $\nu$  ft.<sup>3</sup>/min. in the upper half, and  $p_2, q_2$  the volumes that react to give  $\nu$  ft.<sup>3</sup>/min. in the lower half (see Fig. (3)6).



Then by striking oxygen and carbon balances for each half and eliminating  $\nu$  between the two halves it can be shown that

$$\frac{q_1}{q_2} = \frac{c_1}{c_2} \quad (1)$$

$$\frac{p_1}{p_2} = \frac{(21 - \frac{1}{2}c_1 + (21 - c_1)x/y)}{(21 - \frac{1}{2}c_2 - (21 - c_2)x/y)} \quad (2) \quad . \quad (3/28)$$

where  $c_1, c_2$  refer to the mean percentage of  $\text{CO}_2$  in the upper and lower halves respectively and  $x/y$  is the ratio of  $\text{CO}_2/\text{CO}$  in the original producer gas.

Now suppose for example that the air is introduced above the combustion gas and that there is an excess of air. Then there is naturally a tendency for there to be more air in the upper half of the exit gases and more fully burnt gas in the lower half. Hence in this case  $q_1/q_2$  would be less than unity and  $p_1/p_2$  greater. The more evenly the gas is distributed between the two halves the nearer  $q_1/q_2$  is to 1, and similarly  $p_2/p_1$  measures the evenness of the air distribution. Also, by taking both the ratios  $q_1/q_2$  and  $p_2/p_1$  in conjunction the effect of excess air will be eliminated, because increase in the amount of air ( $p_1 + p_2$ ) makes the air more likely to appear equally in both halves, but the gas less likely, and conversely. The following criteria can accordingly be used and thus evaluation before and after a modification assesses the value of the modification.

Gas distribution index  $= q_1/q_2$ .

Air distribution index  $= p_2/p_1$ .

Mixing index  $= \frac{1}{2}(q_1/q_2 + p_2/p_1)$ .

Recently, Leckie and his co-workers<sup>3.23</sup> have described the use of an experimental furnace for the study of diffusion combustion. The key measurement in his work has been the heat transfer to sections of a subdivided calorimeter on the furnace hearth and he has shown that as far as port designs are concerned those which give the best mixing (see Section 5.7.4) are the best. The combustion conditions are also important, however, the best results

<sup>3.23</sup> LECKIE, A. H., HALL, J. R., and CARLIDGE, C.: "An Experimental Furnace for the Investigation of Open-Hearth Furnace Combustion Problems," Pt. 1, *J.I.S.I.*, **155**, pp. 392-405 (Mar. 1947).

LECKIE, A. H., ALLEN, J. F., and FENTON, G.: *ibid.*, **155**, pp. 405-422 (Mar. 1947).

ALLEN, J. F., COOK, J. H., and FENTON, G.: *ibid.*, Pt. III, **160**, pp. 37-45 (Sept. 1948).

COOK, J. H., and LECKIE, A. H.: *ibid.*, Pt. IV, **160**, pp. 46-56 (Sept. 1948).



### 3.3.4 THE SCIENCE OF FLAMES AND FURNACES

being obtained with the theoretical air supply, high furnace pressures to reduce air infiltration and high calorific value of the fuel.

## 3.4. The Speed of Combustion of Solid Fuel

**3.4.1. General Description of the Processes in Fuel-beds.** When air is passed through a uniformly packed bed of hot solid fuel the oxygen is consumed rapidly and has all disappeared in a distance from the inlet equal to two or three times the mean particle size. At this point the gases sampled with a water-cooled probe of normal diameter are found to contain some 16% of  $\text{CO}_2$  and some 6 or 7% of CO. In practice the bed has always a greater thickness than that corresponding exactly to the complete consumption of oxygen, and in the subsequent layers of the bed  $\text{CO}_2$  is reduced and CO produced with the consumption of a further quantity of carbon and a reduction of the gas and fuel temperature by endothermic reaction.

Three features of this process have been especially studied, namely :

- (i) The variation with distance of the overall interaction of the air stream and the fuel. This variation can be measured by the weight of combustible which has combined with unit N.T.P. volume of ingoing air : a convenient measure of it has been called<sup>3.24</sup> the "relative carbon saturation" (R.C.S.) and is defined as the ratio of this weight to the maximum weight which can combine with this volume. For combustion of all the oxygen in the air to  $\text{CO}_2$  the R.C.S. thus has a value of  $\frac{1}{2}$ , while for combustion of it all to CO it has a value of unity. The variation of the R.C.S. with distance through the fuel-bed is of importance in determining the optimum thickness of the bed for any given purpose.
- (ii) The exact mechanism of the reactions by which oxygen is consumed near the air entry, since the course of these reactions determines the peak temperature reached in the bed and hence ash fusion.
- (iii) The rate of reduction from  $\text{CO}_2$  to CO in the part of the bed in which no oxygen remains. The factors affecting this reaction rate are of great importance in determining the quality of gas made in a gas-producer.

<sup>3.24</sup> THRING, M. W. : "The Degree of Interaction between Air and Solid Fuel : The Effect of Fuel Size," *Coal Research*, p. 70 (Sept. 1944).



**3.4.2. The Relative Carbon Saturation as a Function of Distance Through the Fuel Bed.** In the case of fuels whose combustible matter is largely carbonaceous, e.g. coke or anthracite, the R.C.S. can be calculated from the formula :

$$\text{R.C.S.} = \frac{1 - 0.019\text{CO}_2 - 0.048\text{O}_2}{1 + 0.010(\text{CO}_2 + \text{O}_2)} \quad . \quad . \quad . \quad (3/29)$$

the coefficients being given only to two significant figures, as this type of experiment does not warrant a higher accuracy. It should be noted that according to this formula :

- (1) pure air corresponds to an R.C.S. of zero ;
- (2) combustion gas with the maximum possible  $\text{CO}_2$  content (assumed here to be 21%) corresponds to  $\text{R.C.S.} = 0.5$  ;
- (3) producer gas with the maximum possible CO content (34.6%) corresponds to  $\text{R.C.S.} = 1.0$  ;
- (4) the gas-phase reaction of CO with  $\text{O}_2$  does not change the R.C.S. : for example, a gas consisting of 10%  $\text{CO}_2$ , 9.8% CO and 4.9%  $\text{O}_2$  has the same value as one containing 21%  $\text{CO}_2$  and no CO or  $\text{O}_2$ .

Thus the R.C.S. takes no account of the completeness of purely gas-phase reactions, but represents a true measure of the degree of interaction of the gas and solid fuel.

When the R.C.S. is plotted against  $L$ , the distance from the inlet surface, the curves of Fig. (3)8b are obtained, corresponding to curves of the type shown in Fig. (3)8a for the gas analyses. It will be seen that L. Nichols<sup>3.25</sup> curves, obtained with fuel sized  $\frac{1}{4}$ – $\frac{3}{8}$  inch (average  $\frac{5}{16}$  inch) rise in all cases much more steeply than those of the other workers who used fuel sized 1–1½ inch. When, however,  $L/D$  (where  $D$  is the average particle size) is used as abscissa (Fig. (3)9) the curves of Mott<sup>3.26</sup> and of P. Nicholls<sup>3.27</sup> for cold air come within the spread of those of L. Nichols, showing that the use of this dimensionless abscissa allows, at any rate approximately, for the fuel size. This means also that such curves can be applied with some measure of confidence to fuel-beds with particles outside the size ranges for which the curves have been obtained.

3.25 NICHOLS, L. H. F. : "Gas Sampling and Temperature Measurement in Fuel Beds," *J. Inst. Fuel*, 14, p. 71 (1940).

3.26 MOTT, R. A., and WHEELER, R. V. : "Coke for Blast Furnaces," Iron and Steel Industrial Research Council, *Tech. Report*, No. 1 (Midland Coke Research Committee, 1930).

3.27 NICHOLLS, P. : U.S. Bureau of Mines, *Bull.* 378.



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P. Nicholls and Mott used high-temperature coke, but L. Nichols used anthracite. The limits of experimental error are not shown in Figs. (3)8b and 9, but they probably correspond to about  $\pm 0.05$  units of the R.C.S. up to  $L/D=5$  and  $\pm 0.025$  subsequently.

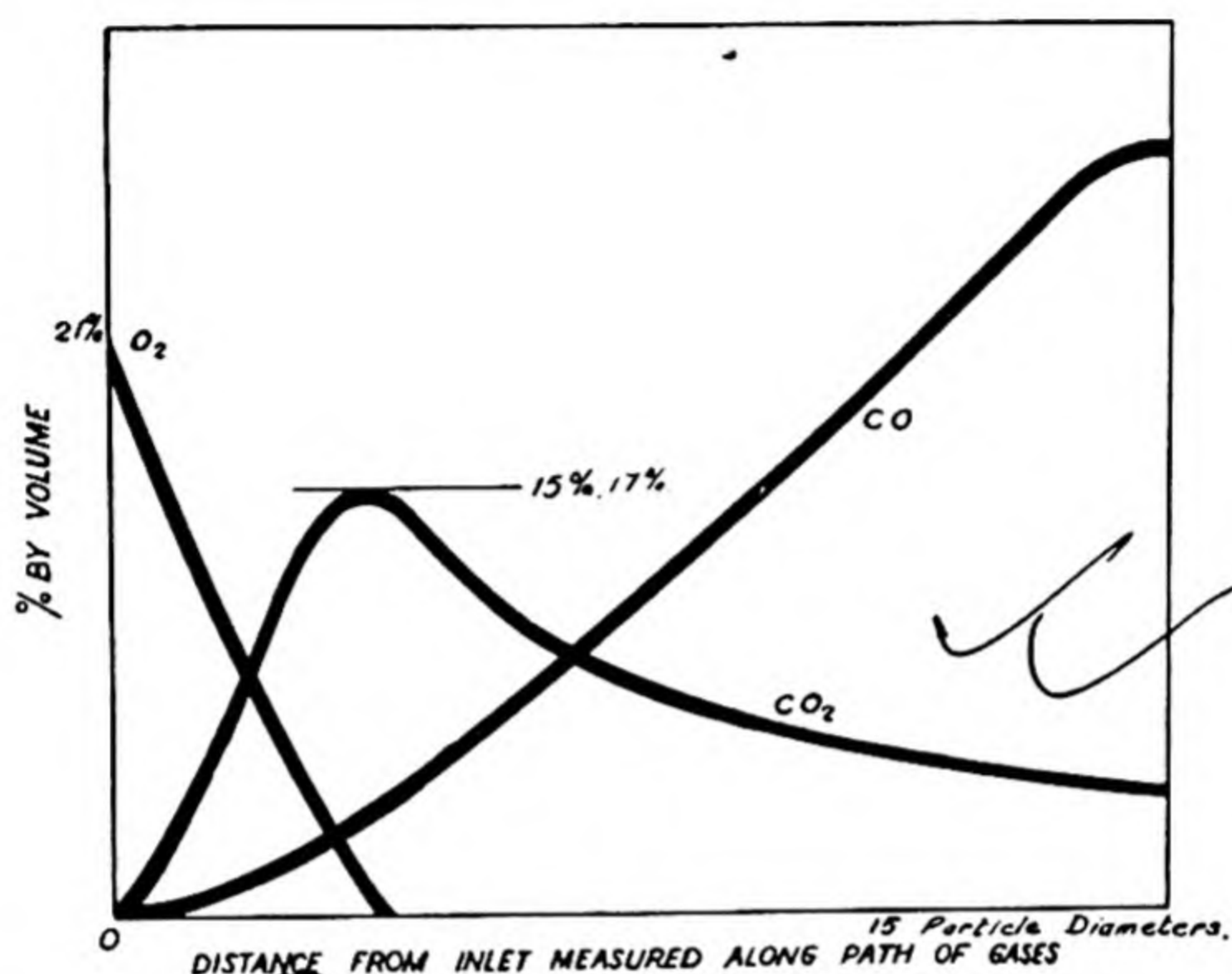


Fig. (3)8a. Typical Gas Analysis in Fuel Bed.

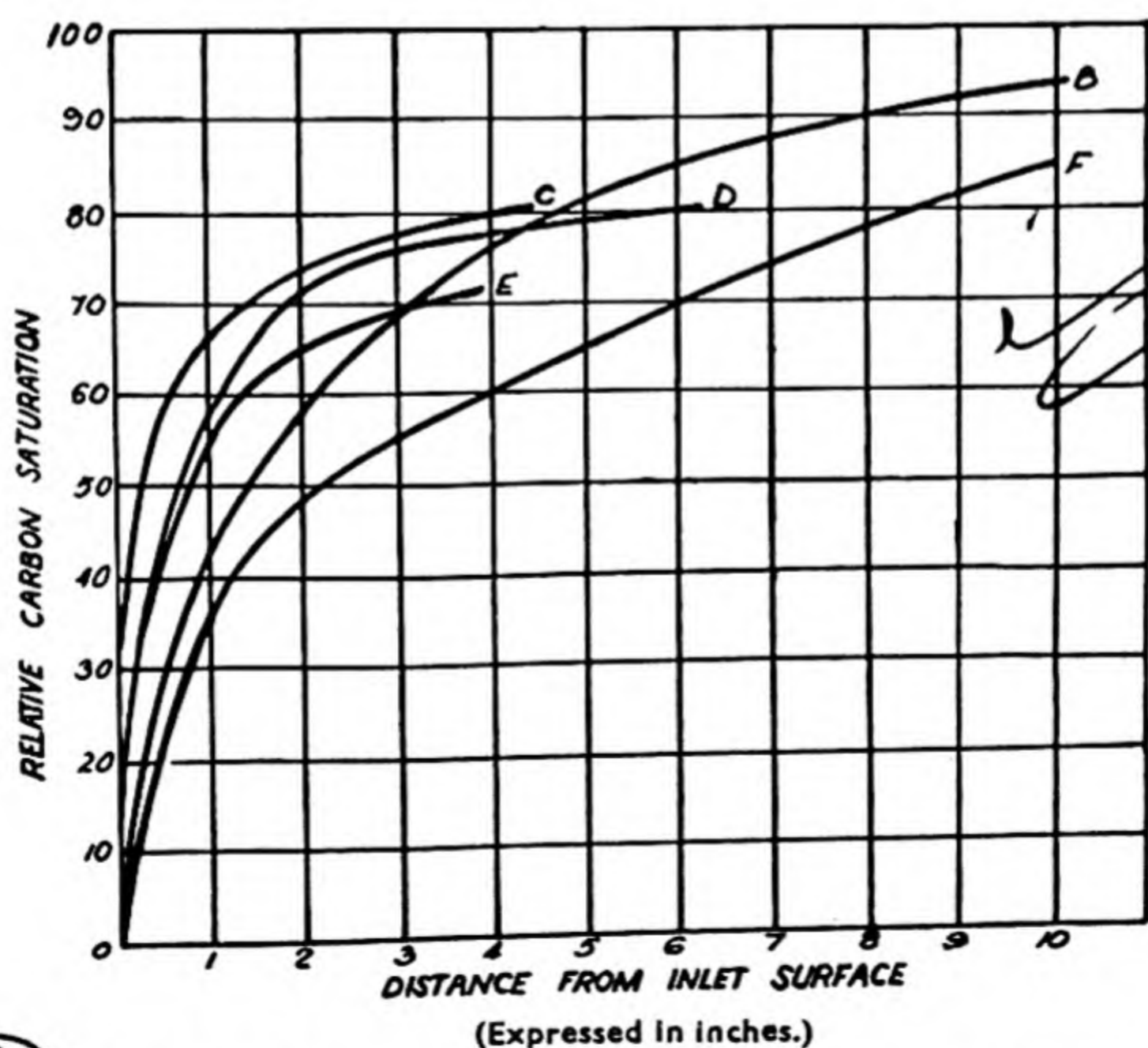


Fig. (3)8b. Relative Carbon Saturation plotted against Distance from Inlet Surface. Fuel Size  $\frac{1}{4}$  to  $\frac{3}{8}$  in. (L. Nichols).

Calculations have been made from the curves of Kreisinger,<sup>3.28</sup> but they show an almost linear rise in the R.C.S. up to 0.80 at  $L/D=8$ . The difference in form of his results from those of the other workers is probably due to channelling, resulting from ash build-up against which insufficient precautions were taken. One point of importance does arise from Kreisinger's curves and this is that an increase of air rate of six times gives no significant change in the slope of the curves.

From these curves the following approximate equations have been calculated and can be applied to the steady-state combus-

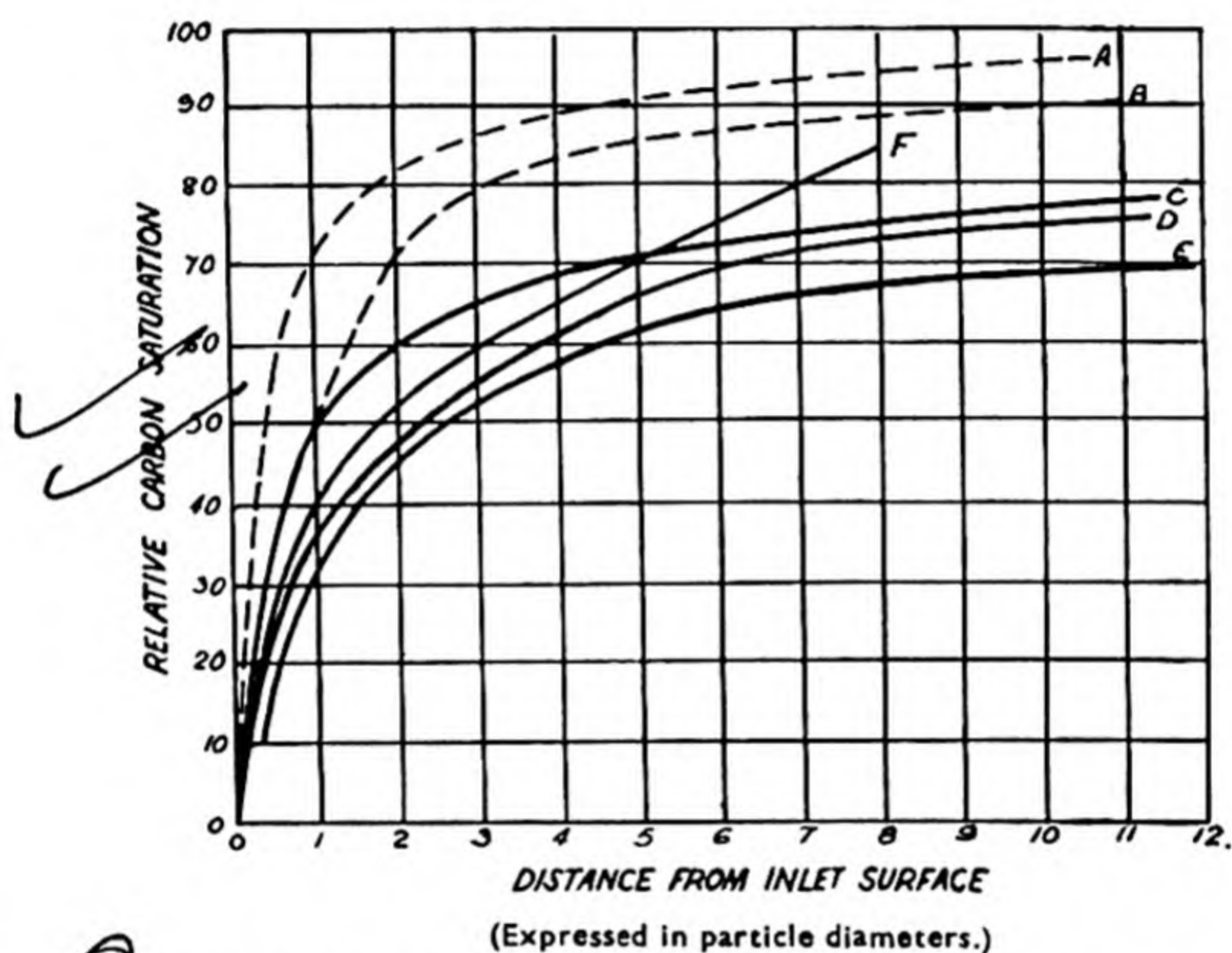


Fig. (3) Relative Carbon Saturation plotted against Ratio of Distance  $L$  from Inlet Surface to Average Particle Size  $D$ .

tion of free-burning fuels under conditions of perfectly uniform packing.

- (1) For cold air the R.C.S. at distance  $L$  from the inlet surface lies between

$$0.90(1 - e^{-\alpha L/D})$$

and

$$0.70(1 - e^{-\alpha L/D}) \quad . \quad . \quad . \quad . \quad (3/30)$$

the former limit applying to cases of low wall-heat loss and reactive fuel, the latter to the opposite conditions ;  $D$  is the

<sup>3.28</sup> KREISINGER, H., OVITZ, F. K., and AUGUSTINE, C. E.: U.S. Bureau of Mines, Paper 137, 1916.



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mean particle diameter measured in the same units as  $L$  and the coefficient  $\alpha$  varies between 0.3 and 0.6, depending on the reactivity and the air rate, being highest when these are high.

- (2) For air preheated to  $300^{\circ}\text{C}$ . the R.C.S. at distance  $L$  from the inlet surface

$$=0.97(1-e^{-\alpha L/D}) \quad . \quad . \quad . \quad . \quad . \quad (3/31)$$

where  $\alpha \doteq 1.0$ .

A practical method of controlling the R.C.S. of the combustion gases by varying the depth of penetration of the air into the bed has recently been described.<sup>3.29</sup> Excess oxygen, an R.C.S. of 0.50 or excess combustible can be obtained by blowing the air down on to a free fuel surface with a velocity which can be controlled independently of the quantity by means of a variable-aperture jet, provided the combustion gases leave through the same free fuel surface. Variation of the jet velocity varies the effective number of particle diameters which the gases contact before they leave. In this way it is unnecessary to supply secondary air, as with ordinary fuel beds, and so very high combustion rates can be obtained with the theoretical  $\text{CO}_2$  content in the gases.

**3.4.3. The Mechanism of Oxygen Consumption in the Early Part of the Fuel Bed.** A great deal of work has been done to ascertain the mechanism of oxidation of carbon and in particular whether  $\text{CO}_2$  or  $\text{CO}$  is the primary product. In 1913 Rhead and Wheeler<sup>3.30</sup> suggested that combustion takes place by the alternate formation and decomposition of a surface compound of carbon and oxygen. When the oxygen was admitted to the fuel particles maintained at a certain temperature  $T$  for a certain time and then the fuel was heated rapidly to  $1000^{\circ}\text{C}$ ., the observed decomposition products varied from practically pure  $\text{CO}_2$  for  $T=200^{\circ}\text{C}$ . to ten times as much  $\text{CO}$  as  $\text{CO}_2$  for  $T=900^{\circ}\text{C}$ . When dry air was drawn steadily through a bed of carbon particles maintained at  $T^{\circ}\text{C}$ . and

<sup>3.29</sup> THRING, M. W.: "A Method for the Control of Combustion Reactions in Fuel Beds," *Trans. Farad. Soc.*, No. 286, **42**, p. 366 (Mar. 1946).

KARTHAUSER, F. B., and SHARPE, G. C. H.: "Downjet Coke Firing for Small Steam Generators." *J. Inst. Fuel*, Jan. 1950.

ROSS, F. R., and SHARPE, G. C. H.: "The Burning of Coke by Downjet." *J. Inst. Fuel*, 1950.

SHARPE, G. C. H.: "The Coke-fired Downjet Furnace in Industry," *J. Inst. Fuel*, 1950.

<sup>3.30</sup> RHEAD, T. F., and WHEELER, R. V.: (1) "The Mode of Combustion of Carbon"; (2) "The Rate of Reduction of Carbon Dioxide by Carbon," *J.C.S.*, **103**, p. 461 (1913); **88**, p. 831 (1912).



the resulting products collected and analysed, the same  $\text{CO}_2/\text{CO}$  ratios were found up to  $550^\circ\text{C}$ . as in the previous experiment but above this temperature it was impossible to avoid secondary gas phase reactions.

Their work provided a basis for modern conceptions of solid-fuel combustion, but the mechanism of the surface absorption of oxygen has been further studied, e.g. by Sihvonen<sup>3.31</sup> and Strickland-Constable<sup>3.32</sup> using the methods of low-pressure combustion of a carbon filament, and study of partially burnt carbon by X-ray diffraction and electrolytic etching. It has been concluded that the surface complex is not a chemical compound (for if it were, given infinite time it would all be decomposed, whereas in fact only a certain fraction can be decomposed at any given temperature), *but is a two-dimensional complex owing its stability to chemi-sorption*, that is to say, the oxygen is held on the surface of the carbon by valency bonds, the carbon atoms being still held in the lattice framework of the solid. This idea is supported by the fact that carbon has an almost vanishingly small vapour pressure even at temperatures over  $2000^\circ\text{C}$ ., which indicates that very strong free valencies exist at the edges of the carbon crystal. There can therefore be a whole set of distinct co-valent compounds of solid carbon and oxygen.

The problems of the decomposition of this complex and hence of the reaction of oxygen with carbon have been studied further by Lothar Meyer<sup>3.33</sup> using electrically heated graphite filaments attacked by oxygen, secondary reactions being avoided by the use of very low pressures. He found that up to  $1500^\circ\text{K}$ . the reaction is first order, the ratio of  $\text{CO}$  to  $\text{CO}_2$  is unity independent of the temperature and pressure (i.e. the reaction can be given by  $4\text{C} + 3\text{O}_2 = 2\text{CO}_2 + 2\text{CO}$ , the activation energy is 20–30 k.cal. and the reaction proceeds by the attack on the basal plane of a single crystal). In the range  $1500^\circ\text{--}1800^\circ\text{K}$ . the reaction is zero order, the ratio of  $\text{CO}/\text{CO}_2$  is 2, but remains independent of  $T$  (equivalent reaction formula  $3\text{C} + 2\text{O}_2 = 1\text{CO}_2 + 2\text{CO}$ ), the activation energy is 70 k.cal. and the crystals are attacked at the edges.

Strickland Constable<sup>3.34</sup> has carried out further work using the techniques of the carbon filament at low pressures and has

<sup>3.31</sup> SIHVONEN: *Trans. Farad. Soc.*, 1938, **34**, 1062.

<sup>3.32</sup> STRICKLAND CONSTABLE: *Fuel*, p. 128 (1944).

<sup>3.33</sup> LOTHAR MEYER: "Chemical Reactions Involving Solids," *Trans. Faraday Soc.*, Symp., p. 1056 (April 1938).

<sup>3.34</sup> STRICKLAND CONSTABLE, R. S.: *Trans. Farad. Soc.*, **40**, p. 333, 1944.



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concluded that the earlier discordant results were due at least in part to secondary effects. Using apparatus in which these secondary effects should not arise he found the primary product to be CO over the temperature range 900–2000° C. Arthur<sup>3.35</sup> has carried out experiments in the temperature range 460–900° C. using varying amounts of PO Cl, which inhibit the gas phase reaction  $\text{CO} + \frac{1}{2}\text{O}_2 \rightarrow \text{CO}_2$  partially or completely. He has concluded that there is a primary reaction forming CO and one forming CO<sub>2</sub> and that their relative rates and hence the ratio of CO/CO<sub>2</sub> in the primary

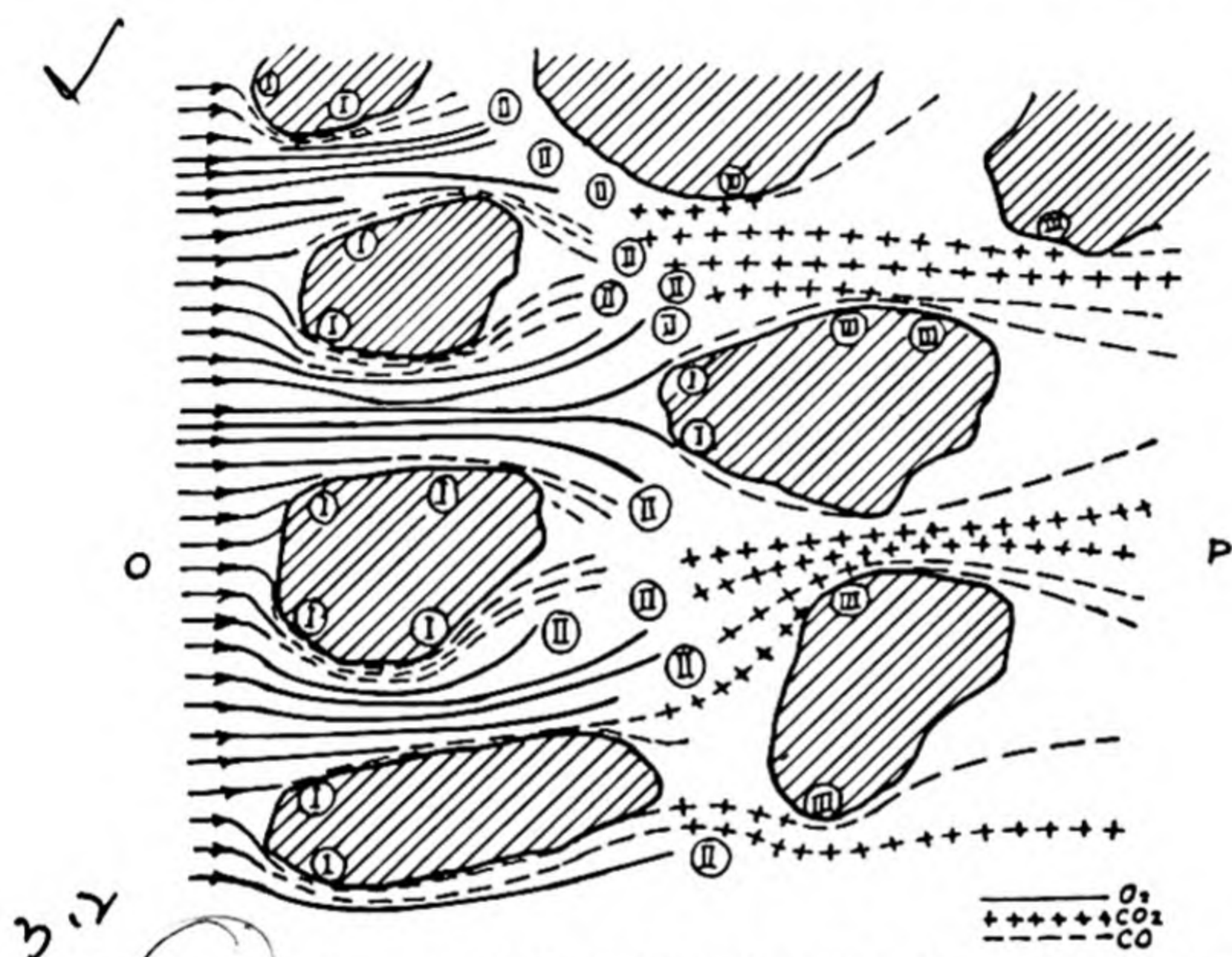


Fig. (3)10. Diagram of the Three Reaction Zones in a Fuel Bed.

products is uniquely determined by temperature, no effect of particle size or nature being found. Thus he finds that the ratio of CO/CO<sub>2</sub> is given by

$$\frac{\text{CO}}{\text{CO}_2} = 10^{3.4} \exp\left(\frac{12400}{KT}\right).$$

A similar use of inhibitors for the temperature range 800–1100° C.<sup>3.36</sup> has indicated that here the CO greatly exceeds CO<sub>2</sub> in the primary products in agreement with Strickland Constable.

It can therefore be concluded that above about 900° C. the

<sup>3.35</sup> ARTHUR, J. R.: *Trans. Farad. Soc.*, **47**, Pt. 2, p. 164, Feb. 1951.

<sup>3.36</sup> ARTHUR, J. R.: *Nature*, **157**, p. 733, 1946.

ARTHUR, J. R., and BOWRING, J. R.: *J. Chem. Soc.* No. 1, p. 51, 1949, Suppl. Issue.



primary product is predominantly CO but below this temperature a mixture is formed.

The great predominance of CO<sub>2</sub> in the gases obtained in the early part of a fuel bed (Fig. (3)8) therefore shows that there must be a considerable amount of gas-phase combustion of CO to CO<sub>2</sub> in the voids between the particles in the early part of the bed. This is known as the "Three-Zone Theory" and is illustrated in Fig. (3)10. It is verified by the fact that the peak CO<sub>2</sub> measured in fuel beds composed of larger-sized particles is appreciably higher than that observed in beds of smaller particles, while the peak CO<sub>2</sub>, which appears in the case of suspension combustion (where the voids very much predominate over the particle surface), is of the order of 20%. A more basic verification of the 3-zone theory is provided by experiments on the effect of removing chain carriers essential for the  $\text{CO} + \frac{1}{2}\text{O}_2 \rightarrow \text{CO}_2$  gas-phase reaction from the reaction zone. Water and other hydrogen-containing materials provide such chain carriers and by using an inhibitor to remove them Arthur<sup>3.36</sup> demonstrated that the CO content of the gases from a laboratory solid carbon combustion apparatus was greatly increased. This indicates clearly that the gas-phase oxidation of CO is a major reaction in fuel beds.

**3.4.4. The Rate of Reduction of CO<sub>2</sub> in Fuel Beds.** Clement, Adams and Haskins<sup>3.37</sup> studied the rate of reduction of CO<sub>2</sub> in beds of carbon arranged in a small tube maintained electrically at a known temperature. They found that the reaction rate  $k = \frac{1}{\text{CO}_2} \cdot \frac{d(\text{CO})}{dt}$  varied between 0.021 at 800° C. and 1.53 at 1100° C. for charcoal, 0.0023 at 900° C. and 1.48 at 1300° C. for coke and 0.12 at 1100° C. and 0.58 at 1300° C. for anthracite, 5-mm. particles being used in all cases. A similar analysis of results on a larger scale has been obtained by Nichols and Thring<sup>3.38</sup> (Fig. (3)11). In this case, however, it appeared that the variation with temperature was not nearly so great as would be expected from the Arrhenius equation, whereas Rhead and Wheeler found good agreement with this, and Clement, Adams and Haskins' results are not far from it. The difference must clearly be due to the fact that the large-scale work analysed by Nichols and Thring applies to temperatures almost entirely above 1100° C., and at this temperature diffusion

<sup>3.37</sup> CLEMENT, ADAMS and HASKINS: U.S. Bur. Mines., *Bull.* 7 (1911).

<sup>3.38</sup> NICHOLS, L. F., and THRING, M. W.: "Physical Processes in a Bed of Fuel," *J. Inst. Fuel* (1940).



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through the stagnant film is as important as the chemical resistance to reaction.\* This is confirmed by the fact that the curve of  $\frac{1}{\text{CO}_2} \cdot \frac{d(\text{CO}_2)}{dx}$  against temperature for results obtained with one

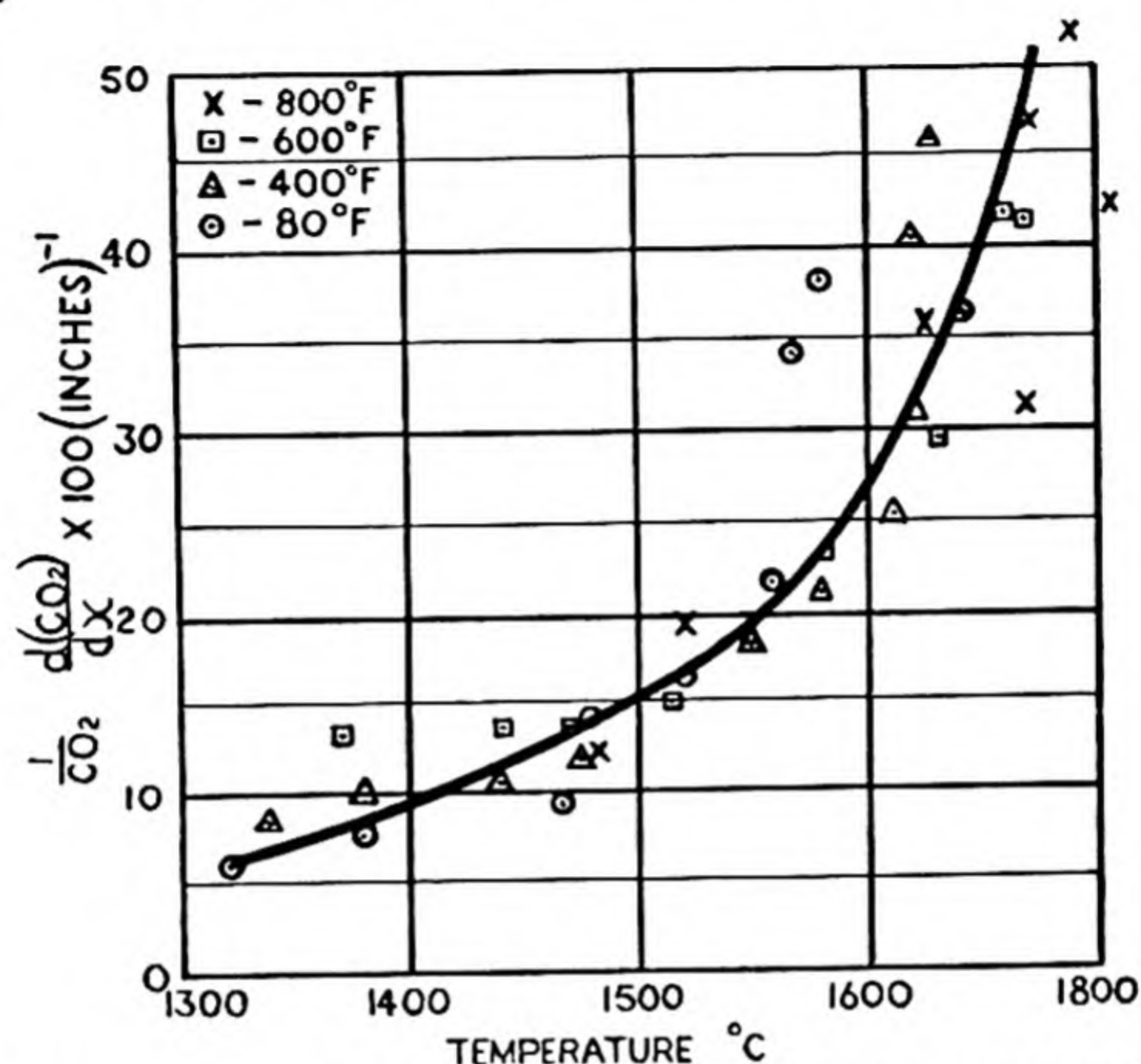


Fig. (3)11. CO<sub>2</sub> reduction rates in a bed of 1-1½ inches coke particles with various air preheat temperatures.

air rate lies considerably above one-third of the height obtained with one-third of the air rate, whereas if the time-rates of the reaction were equal it would lie at exactly one-third. Mayers<sup>3.40</sup> advances an alternative hypothesis to explain the discrepancy based on his explanation of the abnormalities found in comparing results at low pressures with those at atmospheric pressure, which is that

\* In the temperature range 600-900° C. recent work<sup>3.39</sup> has shown that the overall rate of the reaction is not influenced by mass transfer factors and is given by an expression of the form

$$\text{Rate} = \frac{k_1 p_{\text{CO}_2}}{1 + k_2 p_{\text{CO}} + k_3 p_{\text{CO}_2}}$$

CO retards the reaction because it can be adsorbed on the reaction sites.

3.39 GADSBY, J., HINSHELWOOD, C. N., and SYKES, K. W.: *Proc. Roy. Soc. A*, **187**, p. 129, 1946.

GADSBY, J., LONG, F. J., SLEIGHTHOLM, P., and SYKES, K. W.: *ibid.*, **193**, p. 357, 1948.

LONG, F. J., and SYKES, K. W.: *ibid.*, **193**, p. 377, 1948.

3.40 MAYERS, M. A.: "Temperatures and Combustion Rates in Fuel Beds," *Carnegie Inst. Tech. Contribution No. 50*; "Some Factors Affecting Combustion in Fuel Beds," *Carnegie Inst. Tech. Contribution, No. 49*.



they are due to the fact that a great proportion of the reduction actually takes place at the inner surface of the carbon to which the gases diffuse. His evidence is a set of experiments in which he measures the reduction rate for a number of granular beds and monoliths of varying thickness where the gases are passed along the surface. He plots the reaction rate per unit volume against the square of the thickness and concludes that the value obtained by extrapolating to zero thickness is the true reaction rate per unit volume independent of diffusion effects.

In spite of this relatively low variation with temperature compared with that deduced from the Arrhenius equation, however, the variation is sufficiently great for a drop in temperature to play a big role in the final amount of reduction obtained in a fuel bed. Indeed, the main effect on the final  $\text{CO}_2$  of changing the air flow rate lies in the fact that the temperature is higher for the higher flow-rates due to the reduced effect of wall heat losses, which in practical fuel beds more than offsets the relative effects of the rating for constant temperature discussed in the previous paragraph. It is for this reason also that preheat is very effective in increasing the final CO content of the gases, while reactivity of the fuel is particularly important at the lower temperature.

This discussion may be summarised by saying that in a technical gas-producer bed the reduction reaction takes place in a region of temperature where both the chemical resistance (which probably obeys the Arrhenius equation) and the physical resistance to diffusion through the stagnant film surrounding the particles play approximately equal roles. In consequence increase of the gas making rate alters the gas composition because it alters the temperature by changing the cooling due to thermal combustion to the walls. On the other hand fuel reactivity affects the gas composition because it affects the chemical rate.

**3.4.5. The Ignition of Fuel Beds.** Seyler<sup>3.41</sup> and Rosin<sup>3.42</sup> agree that carbon (and coal) and air react with finite speed at all temperatures, the rate of heat liberation being related exponentially to the absolute temperature in a way that agrees with the results of Tu, Davis and Hottel.<sup>3.43</sup> Since the heat release depends directly on

<sup>3.41</sup> SEYLER, C., and JENKINS, T. E. : Symposium on "Ignition of Fuels on Grates," *J. Inst. Fuel*, **11**, p. 122 (1938).

<sup>3.42</sup> ROSIN, P. O., and FEHLING, H. R. : "The Ignition of Coal on a Grate," *J. Inst. Fuel*, **11**, p. 102 (1938).

<sup>3.43</sup> TU, C. M., DAVIS, W., and HOTTEL, H. C. : *Ind. Eng. Chem.*, **26**, p. 749 (1934).



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the heat of reaction, if the temperature of the system is gradually raised it will eventually become self-heating at the point when the rate of reaction is sufficient to supply the heat losses, which go up according to a lower power of the temperature. Rosin<sup>3.42</sup> shows that when fuel on a moving grate is heated from above while the air comes from below, beds of smaller particles may ignite faster or slower than those of large particles, since there are two opposing effects on size. On the one hand the poorer heat transfer by conduction and radiation of the smaller particles delays their ignition as compared with the larger ones and offsets the greater surface available for convection and reaction. Bennett<sup>3.44</sup> concludes that there is no such thing as an absolute ignition temperature independent of the apparatus, whereas Seyler considers that there may be a critical temperature independent of the characteristics of the heat loss system at which gaseous products ignite in the voids. Both workers find a fanning action of the blast on the fuel which results in more rapid ignition at higher air rates. Grumell and Dunningham<sup>3.45</sup> find, with the others, that ignition is very sensitive to the reactivity of the fuel while the effect of the size is complicated by two opposing factors.

Mayers<sup>3.40</sup> has shown that the ignition rate of a fuel depends primarily on its reactivity (which probably depends on the micropores<sup>3.46</sup>) whereas the rates of reaction of  $O_2$  and  $CO_2$  with the fuel in a bed at practical temperatures ( $1000^\circ$ – $1500^\circ$  C.) depends primarily on the porosity of the coke (macro-pores) and on the coke size. He reaches these conclusions by analysing curves of gas analysis against distance through the bed, obtained with a number of cokes made under different conditions from the same coal.

**3.4.6. Water-gas Reactions.** From the energy-release point of view the fuel bed in a water-gas apparatus may be compared to the reducing zone in a gas-producer. In each case endothermic fuel-consuming reactions are proceeding ( $CO_2 + C = 2CO$ —40,800 C.H.U. in the gas-producer,  $C + H_2O \rightarrow CO + H_2$ —32,300 C.H.U. in the water-gas apparatus) which absorb heat and raise the combustible content of the gas. Hence, in a water-gas apparatus, just as in the gas-producer, the extent to which these reactions proceed depends predominantly on (a) the fuel reactivity, (b) the amount

3.44 BENNETT, J. G.: "Discussion on Ignition Symposium," *J. Inst. Fuel*, **11**, p. 218 (1938).

3.45 DUNNINGHAM, A. C., and GRUMMELL, E. S.: "Ignition of Fuel on Grates," *J. Inst. Fuel Symp.*, **11**, pp. 117 and 129 (1938).

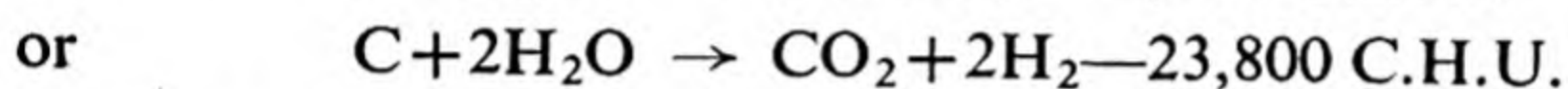
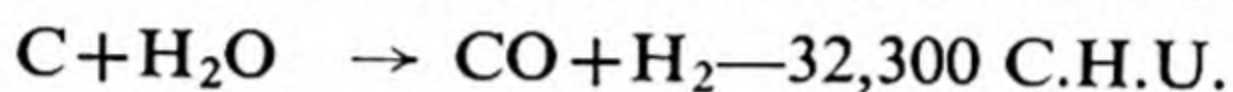
3.46 B.C.U.R.A. Conference on the Ultra-fine Structure of Coal and Coke.



and uniformity of the gas contact with the fuel and (c) the prevailing temperatures. In each case there is no question of equilibrium being reached in these gas-solid reactions until they have quenched themselves by absorbing all the available energy. At lower temperatures where chemical reaction rates predominate over mass transfer considerations the overall reaction rate is given<sup>3.39</sup> by the expression

$$\text{Rate} = \frac{k_1 p_{\text{H}_2\text{O}}}{1 + k_2 p_{\text{H}_2\text{O}}}$$

Chemically, water-gas formation is more a complicated process than that which occurs in the gas-producer, as it involves two competing combustible products CO and H<sub>2</sub>. The question whether the primary reaction between steam and carbon is



has not yet been finally settled. Warner<sup>3.47</sup> and Dolch<sup>3.48</sup> incline to the view that the former is primary, on the ground that the tendency is for measured values of the water-gas constant

$$K = \frac{(\text{CO}) \cdot (\text{H}_2\text{O})}{(\text{CO}_2) \cdot (\text{H}_2)} \quad \begin{array}{l} \text{(expressed for the} \\ \text{reaction ending with} \\ \text{H}_2\text{O vapour)} \end{array}$$

to be higher than the equilibrium value corresponding to the measured temperature. Scott,<sup>3.49</sup> on the other hand, considers the second reaction to be certainly primary in the case of lignite char at temperatures below 1000° C., as shown by the fact that the gas tends to the composition CO<sub>2</sub> 33%, H<sub>2</sub> 67% as the contact time is decreased. The most recent work indicates that CO is the primary product.<sup>3.39</sup>

There is, at all events, a strong tendency for the equilibrium value of  $K$  to be reached for the gas-phase reaction  $\text{CO} + \text{H}_2\text{O} \rightleftharpoons \text{CO}_2 + \text{H}_2$ , presumably by reaction in the voids between the particles, particularly at higher temperatures. As  $K$  decreases with decrease of temperature (e.g. 0.72 at 1000° K., 1.41 at 1200° K.) the highest H/CO ratio can be obtained if the equilibrium is attainable at a lower temperature. Hence, synthesis gas (which has 2 vols. of

<sup>3.47</sup> WARNER, B. R. : "Mechanism of the Steam-carbon Reaction," *J. Amer. Chem. Soc.*, **65**, p. 1447 (1943).

<sup>3.48</sup> DOLCH : "Das Wassergas" (Springer, Berlin).

<sup>3.49</sup> SCOTT, G. S. : "Mechanism of the Steam-carbon Reaction," *Ind. Eng. Chem.*, **33**, No. 10, p. 1279 (1941).



### 3.4.6 THE SCIENCE OF FLAMES AND FURNACES

H<sub>2</sub> to 1 of CO) requires a reactive fuel and a low bed temperature; whereas if calorific value is the primary objective, as in a gas-works water-gas plant, high temperatures provide the best operation.

**3.4.7. Pulverised Fuel Firing.** In pulverised fuel firing, unlike fuel-bed firing, the individual particles may be treated separately, since each is completely surrounded by a gas layer and traverses the same path as the reacting gas. In this case, therefore, it is possible to reach the peak CO<sub>2</sub> without subsequent reduction to CO, so that combustion, once ignition has been reached, is a simple process whose ultimate limit is set only by depletion of the O<sub>2</sub> and carbon and does not rise to a peak, fall again and then have to be re-raised, like fuel-bed combustion. Here again, however, the rate of reaction is jointly determined by the resistance to diffusion of gas molecules to and from the surface and the chemical resistance, which probably obeys the Arrhenius equation.

Experiments carried out on single particles large enough to be individually weighed have proved of considerable value in studying the relative importance of these two resistances at different temperatures. Tu, Davis and Hottel<sup>3.43</sup>, working with 1-inch spheres of pure carbon, obtained reasonable agreement between the theory that the rate of reaction is governed by the sum of these two resistances and the measured rate of burning. When they plotted the logarithm of the reaction rate against the reciprocal of the absolute temperature, they found a linear variation in the region corresponding to low temperatures where chemical resistance is the controlling factor, the line being the same for all gas velocities. Over the region corresponding to higher temperatures the logarithm of the reaction rate fell below the line, the exact position in this case depending appreciably on the velocity. They found that the reaction rate was proportional to the oxygen concentration, a fact which implies that the reaction of the first order has the practical consequence that a certain percentage of excess air is necessary if combustion is to be carried to completion in a finite volume.

The weakest link in the theoretical argument discussed in this paper was that part concerned with the variation of the stagnant film thickness with velocity. An experiment was therefore devised<sup>3.50</sup> in which a stagnant film was produced by placing a flat disc of carbon at the bottom of a cylindrical cup and then passing a gas stream across the rim of the cup. According to the theory the

<sup>3.50</sup> DAVIS and HOTTEL: "Combustion Rate of Carbon," *Ind. Eng. Chem.*, **26**, p. 889 (1934).



reaction rate should be inversely proportional to the sum of the stagnant film thickness and a thickness corresponding to the chemical resistance. Such a variation was in fact found provided the gas velocity did not exceed a certain amount. They concluded that the stagnant film was in fact about three times thinner than the value which would be expected from similarity conditions, but varied in the expected way with gas velocity. Parker and Hottel<sup>3.51</sup> measured the gas composition at various small distances from the surface of a suspended carbon block and concluded that the oxygen does not approach zero even at the surface of the particles except at high temperatures. Their conclusions as to the absence of appreciable quantities of CO may be vitiated by the fact that they did not quench their gas samples: surface combustion may have taken place in their silica sampling tubes.

In the case of pulverised coal the temperature is high and the diameter of the particle small, so that the equation calculated from the stagnant film theory reduces to the simple form

$$10^3 K_p = \frac{0.0109 p_{O_2} T_p^{0.5}}{d} \quad \dots \quad (3/32)$$

where  $K_p$  is the reaction rate expressed in gm./sec. cm.<sup>2</sup> ;

$p_{O_2}$  is the partial pressure of oxygen at the point concerned (atm.) ;

$T_p$  is the temperature of the particle (° K.) ; and

$d$  is its diameter (cm.),

which on integration gives

$$t_b = \frac{7.3 \times 10^4}{\epsilon^{\frac{1}{3}}} \cdot \frac{\gamma d_o^2 (1 + \epsilon)}{T_p^{0.5}} \left\{ \frac{1}{2} \ln \frac{\epsilon^{\frac{2}{3}} - \epsilon^{\frac{1}{2}} + 1}{(\epsilon^{\frac{1}{3}} + 1)^2} \sqrt{3} \left( \frac{\tan^{-1} 2\epsilon^{-1/3} - 1}{\sqrt{3}} + \frac{\pi}{6} \right) \right\} \quad \dots \quad (3/33)$$

for the burning time  $t_b$  of the particle, where

$\gamma$  = density of particle gm./cm.<sup>3</sup>,

$\epsilon$  = fraction of excess air,

$d_o$  = initial particle diameter, cm.

It should, however, be pointed out that in applying the final formula of Parker and Hottel to the combustion of pulverised coal,

<sup>3.51</sup> PARKER and HOTTEL: "Combustion Rate of Carbon," *Ind. Eng. Chem.*, **28**, p. 1334 (1936).



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no account is taken of such possibilities as the swelling up of the coal particle observed by Sinnatt, "the cenosphere," nor of the effect of volatile matter. The latter certainly burns in an envelope around the particles and thus affects the combustion of the particles. This is indicated by the observation of Smith and Gudmundsen<sup>3.52</sup> that a comparatively small amount of water vapour materially affects combustion.

Two sets of experiments on the actual combustion times of coal dust are recorded, those of Godbert and those of Griffin, Adams and Smith. In each case the particles in the furnace were photographed on a rotating drum by their own light. Godbert found that the combustion time was roughly proportional to the specific surface and could be correlated with the reactivity found by oxidising for 3 hours in air at 100° C. and measuring the volume of O<sub>2</sub> taken up by the fuel, but it should be noted that the tracks on the photographs correspond only to the first stage of combustion.

The experiments of Griffin, Adams and Smith resulted in the discovery of a negative temperature coefficient, i.e. the particles took longer to burn at higher temperatures. This has been variously explained as being due to the formation of less reactive coke at higher temperatures or to the smaller number of oxygen particles diffusing to the surface, but it may well be due to the fact that they measured the time of emission of light rather than the actual combustion time.

As regards the effect of particle size on the time of combustion, Burke and Schumann<sup>3.18</sup> conclude that the time is approximately proportional to the square of the particle size, while Grebel states that it is proportional to the particle size.

It is not possible to say finally from the work on this subject that a fully checked formula representing the combustion time of coal particles as a function of the variables is available, and in all probability even if it were available it would have to be revised for each new coal. A formula along the lines of that of Parker and Hottel, taking into account coking, combustion of the volatiles and the formation of cenospheres, might be developed. The fact that such a formula is not available shows that it cannot really be said that the mechanism by which pulverised coal burns is understood, even to the extent diffusion flames and Bunsen flames are understood. Semi-empirical formulæ similar to that of Heiligenstadt for gaseous

<sup>3.52</sup> SMITH, D. F., and GUDMUNDSEN, A.: *Fuel in Science and Practice*, 11, p. 124 (1932).



combustion are more likely to be of practical value to furnace builders. The best of these so far available is that of Rosin.<sup>3.53</sup>

### 3.5. The Speed of Combustion of Liquid Fuels

**3.5.1. Oil-firing Methods.** Very broadly it may be said that oil is fired for heating purposes in two main ways: it can be substantially evaporated before it ignites so that it burns as a vapour, or it can be broken up into fine droplets which are heated mainly by radiation as they mix with air, so that the droplets are evaporating while combustion is proceeding. The first method is in essence the same as the methods of combustion of gases and need not be treated separately here, the only special features being the methods by which heat from the flame is transferred to the incoming oil to evaporate it. These may be the heating of a tube in the flame through which the oil passes, radiation and conduction from the flame to oil in an open vessel or spraying of oil on to a surface heated by radiation from the flame or even the use of a wick (see e.g. Heiple and Sullivan, "Mechanisms of Combustion, Relation to Oil Burner Design," *Trans. A.S.M.E.*, p. 343, May 1948). Vaporising burners are used in small domestic boilers, but are not applied to industrial furnaces to any significant extent; they must be used with oils which vaporise at a temperature lower than that at which they decompose and hence are not suitable for the cheaper heavy fuel oils.

The combustion of oil by means of burners which break it up into fine droplets is usually referred to as "atomisation," but in fact the droplets range from 10 to 200 $\mu$  and in some cases some much larger droplets are also present. A good commercial atomiser produces perhaps 85% of its droplets under 50 $\mu$  and they cover a fairly wide size range. The remainder of this section is devoted to the combustion of "atomised" oil.

**3.5.2. Methods of "Atomising."** The three main methods of producing fine oil droplets for burning heavy oils in the liquid state are:

- (1) the centrifugal burner in which a mechanically rotated disc or cup produces droplets by throwing off the oil from its rim;
- (2) "pressure atomisers" in which the fuel is supplied under pressure through a fine orifice of appropriate shape; and
- (3) injector or blast atomisers in which a high velocity stream of steam or compressed air turns the oil into droplets.

<sup>3.53</sup> ROSIN, P. O.: International Conference on Bituminous Coal, 1933.



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The centrifugal burner requires a high-speed electric motor or compressed-air turbine to drive it and the mean droplet size produced is a function of the speed, speeds of 10,000 r.p.m. and over being necessary to produce the bulk of the droplets less than  $50\mu$ . The droplet size is, however, relatively independent of the rate of fuel supply, so that this system is valuable when a wide range of combustion rates is needed. This burner gives a spray which spreads out almost at right angles to the axis of the spinner, which is usually parallel to the flow of the surrounding combustion air. The burner is thus more suitable for boilers than for furnaces, since it is liable to give impingement of burning oil on the surrounding walls.

Pressure atomisers are used also mainly in boilers and in gas-turbine combustion chambers, and they give a hollow cone of droplets with a fairly wide included angle of  $60^{\circ}$ – $80^{\circ}$ , so that the shape of the flame is intermediate between the other two. The fuel enters a hollow cylindrical or conical chamber tangentially and leaves through a hole on the axis so that it emerges with a very rapid rotation resulting from the approximate conservation of angular momentum.<sup>3.54</sup> This type of atomisation gives the greatest variation of drop size when the oil throughput is altered. This is because the pressure drop of the oil in flowing through the nozzle is proportional to the square of the throughput and the pressure must exceed a certain minimum figure to obtain satisfactory atomisation.

Burners used for oil firing in the open-hearth furnace are mainly steam atomised, the atomising being effected by impacting the steam at a high velocity on the oil as the latter issues from a small tube inside the steam tube or from an annulus outside it. The resulting mixture of oil droplets and steam then either leaves the burner mouth or, in many designs, has to flow down a long tube to reach the burner mouth, in which case it is probable that considerable reaggregation of droplets into larger ones takes place. No method of measuring the droplet size from such burners, which supply up to 500 gals./hr. of oil, has yet been developed, although smaller burners have been tested by spraying molten paraffin wax and sieving the frozen droplets at room temperature<sup>3.55</sup> and by spraying oil and freezing the droplets by catching them on a plate

3.54 GREEN, H. L.: "Problems in the Atomisation of Liquids," Institute of Physics Conference on Fluid Flow, 1951.

3.55 JOYCE, J.: "The Wax Method of Spray Particle-size Measurement," Shell Petroleum Co., Ltd., I.C.T. Report No. 7.



cooled by solid  $\text{CO}_2$ .<sup>3.56</sup> The development of these apparatuses on a larger scale or, better still, of methods by which the droplets from an actual operating burner can be photographed or sampled, will provide the only scientific methods for improving the design of such atomising systems.

These fluid atomised burners can be operated over a fairly wide range, e.g. 4 : 1, and require 8–2 lb. of steam per gallon of oil. There is little doubt that air atomisation gives a higher temperature flame and probably a shorter one so that steam atomisation and the dilution of the flame with the steam is a very unsatisfactory process for high-temperature furnaces from the thermodynamic point of view. It is likely, therefore, that the use of air atomisation will increase in spite of the greater ease of supplying steam.

### 3.5.3. The Factors Governing Combustion of "Atomised" Oil.

When the oil droplets leave the burner mouth they receive heat by convection from preheated combustion air and recycled combustion products and by radiation back from the subsequent flame and from the walls of a furnace. In furnaces the wall radiation is very large so that less attention has to be paid to achieving good atomisation and rapid mixing for furnace firing than for boiler firing, but it is certain that even in the open-hearth furnace very large droplets which fall out of the flame lead to fuel wastage and often to undesirable effects on the charge and brickwork also.

The heat reaching the droplet causes evaporation and cracking, and it is probable that the droplet burns mainly by gas-phase reactions between a cloud of vapour surrounding it and air which diffuses into this cloud and by molecular diffusion after being transported to the surface of it by eddy diffusion in the manner discussed above in Section 3.4 on diffusion combustion of gases and in 3.5.4. below. The luminosity of an oil flame is, unlike the pulverised coal flame, due to the carbon formed in the diffusion combustion of these evaporated hydrocarbons and not to the hot droplets, since these will not heat up above the evaporation temperature until all the material which can be evaporated has done so and only a cracked residuum remains. When the oil droplets exceed  $50\mu$  the residuum after completion of evaporation is a swelled-up carbon bubble called a cenosphere, similar to those found by Sinnatt in the combustion of pulverised coal. The actual relative speeds of cracking and evaporation will depend on the rate

<sup>3.56</sup> HOTTEL, H. C. : "Spray and Particle-size Investigations carried out at M.I.T." (Private communication).



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of receiving heat by radiation as well as on the droplet size and the oil composition, since a very high temperature in the surroundings will increase the temperature gradient through the particle itself. It is possible that in the open-hearth furnace the droplets actually explode, especially oils with two distinct boiling ranges, whereas those with a wide, uniform boiling range are more likely to evaporate smoothly. All these processes require a great deal of further investigation, for which the droplet sampler discussed in Section 3.5.2 will be of great value if it can be developed.

Another method of study which is being fruitfully pursued, more particularly on the lighter hydrocarbons, is the investigation of the combustion of a single suspended droplet or of droplets falling through an electric furnace.<sup>3.57</sup> In the former case the droplets are larger than those occurring in flames and the remote surroundings have been mostly cold, so that evaporation is caused by heat transfer from the hot combustion zone around the particle. Nevertheless this method does make possible a very detailed survey of the combustion and evaporation processes, just as the work on suspended carbon spheres discussed in Section 3.4.7 throws light on the combustion of pulverised coal. Thus Godsave has shown that the burning drops evaporate at a rate proportional to the first power of the radius, confirming a theoretical study of evaporation made by Probert<sup>3.58</sup> that the vapour pressure of the particular hydrocarbon used is not a major factor, the rate of evaporation depending primarily on the rate of flow of heat to the droplet and the heat required to raise the temperature of the fuel to its boiling-point and to evaporate it. It will be very interesting to see these results extended to the supply of heat by radiation from hot walls, to smaller droplets, and to mixtures of heavier hydrocarbons with various ranges of boiling-points and various cracking tendencies.

The other technique which has been used to study the combustion of individual droplets is to let them fall through an electric furnace and either photograph the visible trace, as Smith and Gudmundsen<sup>3.52</sup> did for powdered coal, or quench the droplets after various exposure times and find what remains.

3.57 GODSAVE, G. A. E.: "Combustion of Droplets in a Fuel Spray," *Nature*, **164**, p. 708 (22 Oct., 1949).

SPALDING, D. B.: "Combustion of Liquid Fuels," *Nature*, **165**, p. 160 (28 Jan., 1950).

3.58 PROBERT: "The Influence of Spray Particle-size and Distribution in the Combustion of Oil Droplets," *Phil. Mag.*, **37**, p. 94 (1946).



**3.5.4. Air-fuel mixing in oil combustion.** In addition to classifying oil burners according to the method of atomising the oil, it is necessary also to define them by the manner in which the combustion air is mixed with the stream of evaporating droplets. Here the distinction is between those in which the bulk of the combustion air is entrained into the stream by the momentum of the fuel jet and those in which the combustion air is made to move and the fuel stream is sprayed across the air. The former are typified by the injector burners used in open-hearth furnaces which depend on the atomising air or steam to give the fuel jet a very high velocity (of the order of 500–1000 ft./sec.) and momentum. Collins<sup>3.59</sup> has shown that with these burners the combustion length for the oil is primarily governed by the rate of entrainment of the air, i.e. mixing is the slowest process, so that the flame length can be predicted from mixing model experiments. These burners give a long narrow flame.

The other type of mixing is that produced by what may be called proper burners in the sense that they fix the whole mixing pattern independently of furnace walls. It includes pressure and centrifugal atomisation burners where the fuel droplets are sprayed across the air stream and blast atomisers in which 40–100% of the combustion air is used with a velocity of 300 ft./sec. to atomise the fuel (in these burners the air has a high rotary velocity so that the flame is wide angle). With these burners mixing is so rapid that the evaporation and combustion of the oil droplets take times comparable with or greater than the aerodynamic mixing time.

## 3.6. The Control of Combustion

**3.6.1. The Three Aspects of Control.** In this monograph combustion is regarded as a process of energy conversion, hence the success of the process depends on the control of the conversion process which can be obtained. There are three essential forms of control, namely :

- (1) the control of the quantity of energy released in the whole appliance, expressed in C.H.U./hr. ;
- (2) the control of the quality of the energy released as measured, for example, by the theoretical flame temperature which implies control of the degree of completeness of the final reactions and of the excess air;

<sup>3.59</sup> COLLINS, R. D.: "Application of the Momentum Equation to the Oil Flame." *BISRA Report*, SM/A/169/49.



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- (3) control of the course of combustion, i.e. control of the flame length or heat release expressed in C.H.U./unit volume within the appliance.

In each case control may mean merely that the factor is maintained at or near a desired value or that it can be maintained at a

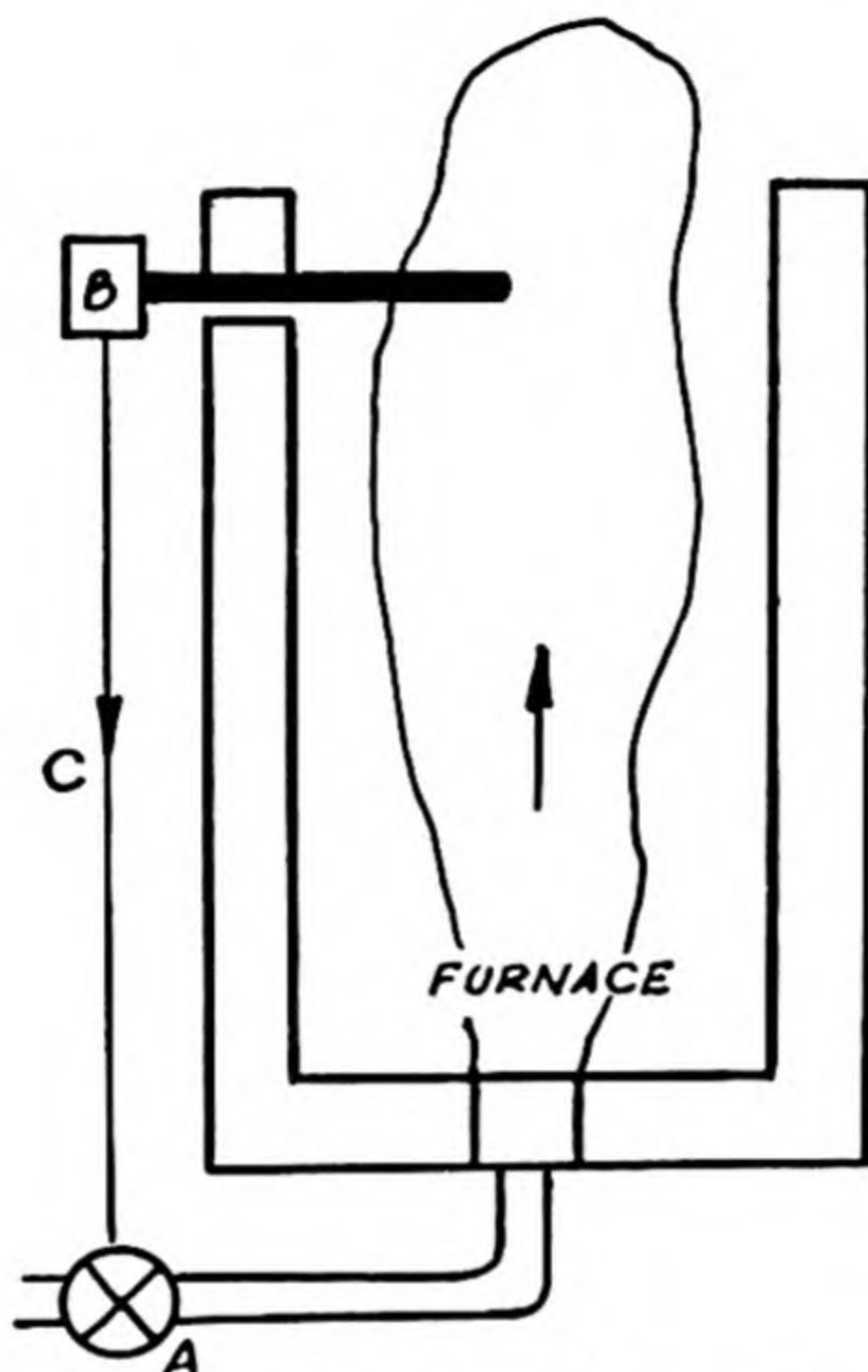


Fig. (3)12. Diagram of General Control System for a Single Variable.

number of values over a more or less wide range. In the former case, however, the fact that the appliance must give good control of the factor concerned is not always clearly recognised because no special precautions are taken to achieve this control.

A diagram of a general control system is shown in Fig. (3)12. Such a system has to be applied to each of the types of control. It consists of the mechanism *A* by which the combustion factor concerned can be controlled ; the sensitive apparatus *B* for measuring the effect of this factor on the furnace system, i.e. for measuring how closely the desired aim has been achieved and, finally, the linkage *C* by which the observations of *B* react on *A*. The effect measured by *B* will be called the "observed variable." It may be simply observed by eye or by an observer with an instrument ; in each of these cases the link *C* by which the observed variable *B* controls the



mechanism *A* involves the human operator of the furnace. In automatic control the sensitive apparatus *B* which observes the observed variable is linked by a mechanical device to the mechanism *A*, and it is only necessary for the furnace operator to set the device so as to give the desired value for the variable.

**3.6.2. Control of the Quantity of Heat Released by Combustion.** This is the simplest and most obvious form of control, corresponding to blowing a fire to get a fierce heat. The quantity of heat released may be controlled, as in the case of solid-fuel combustion, by varying the amount of air drawn through the bed or by varying the amount of fuel, such as gas or oil, supplied to a furnace.

This aspect of control can never be regarded as completely independent of the second one considered in Section 3.6.3, since it is always necessary to control the amount of energy released by varying both the air supplied and the fuel consumed, so that as far as the mechanism of control is concerned the two are always interconnected. On the other hand, the results of control of the two are quite different, and hence the observed variable which should be linked up by an automatic mechanism to control each is quite different. Thus the amount of heat released should be controlled ideally by the rate of heat supplied to the charge, or the furnace or charge temperature. The object of control of the quantity of heat supplied is therefore primarily to ensure that the furnace heats the material at the required rate or to the required temperature.

**3.6.3. Control of the Quality of Combustion.** In this case the object is primarily to ensure that the furnace performs its task with a minimum fuel consumption. The mechanism for controlling the quality of combustion is usually provided by variation of the fuel-air ratio, although in the case of solid fuel combustion it is necessary to control the degree of interaction between the air and fuel and then maintain the requisite fuel-bed size by varying the rate of fuel feed. The observed variable which indicates how well the quality of combustion is being controlled is preferably the composition of the gases leaving the furnace which in most cases should contain a minimum of both unburnt combustible and excess oxygen and hence a maximum  $\text{CO}_2$ . The relations between these two forms of control in an ideal system are indicated in Fig. (3)13, which is appropriate for gas firing. The pair of valves *a* and *b* control the quantities of gas and air respectively. They are linked together by the link *f*, and when this is in any one position the quantities of fuel and air remain in a constant proportion.



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This pair of valves is operated by the control mechanism *c* from a temperature or heat flow indicating device *d* in the furnace chamber. A gas analysis indicator *e* operating on samples withdrawn from the exit gases varies, by means of the mechanism *f*, the ratio of air and gas at any given position of the gas valve.

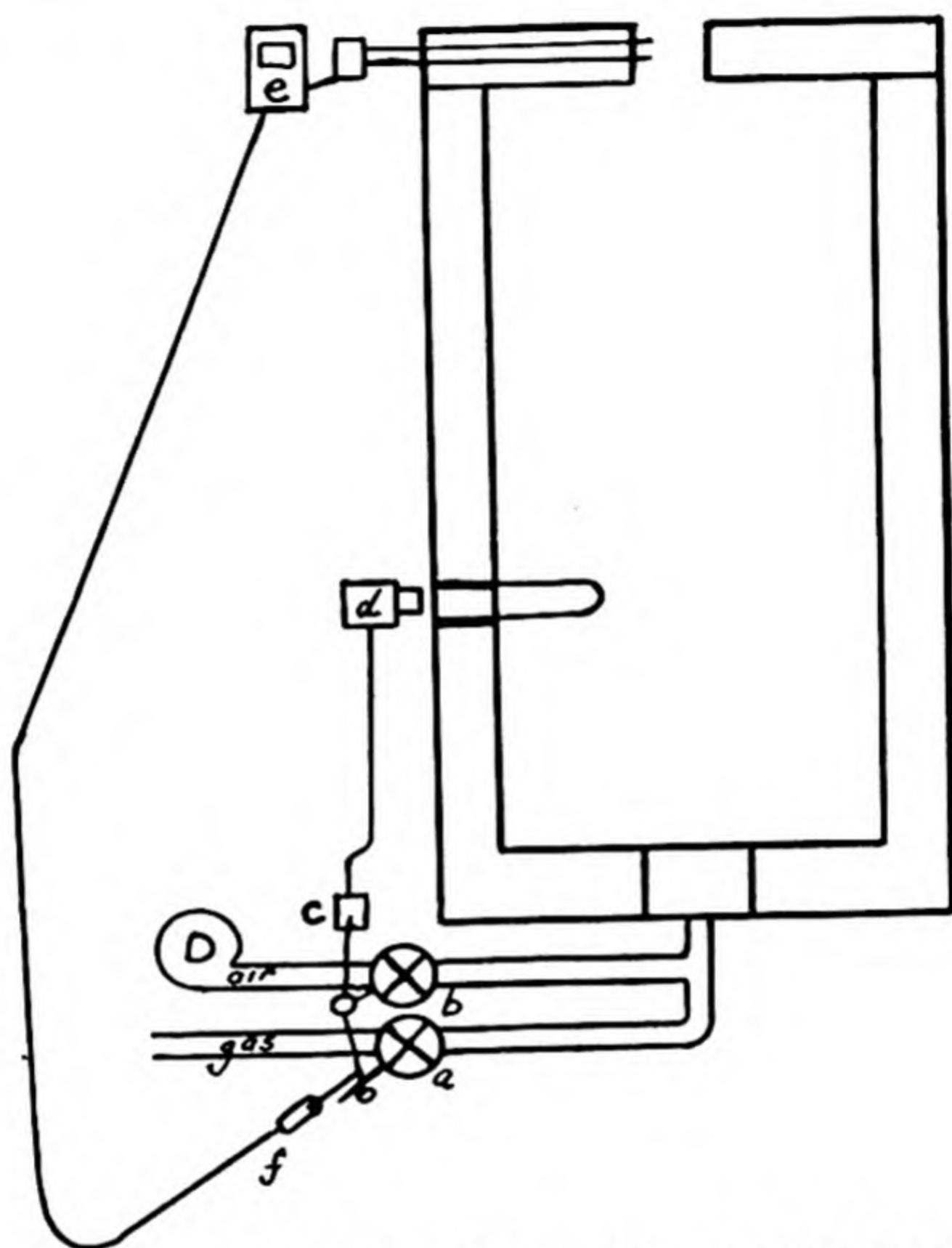


Fig. (3)13. Simultaneous Control of Rate of Heat Liberation (Quantity of Combustion) and of Exit Gas Composition (Quality of Combustion).

**3.6.4. Control of the Course of Combustion.** For optimum fuel economy it is nearly always desirable to have the flame as short as possible in order that the temperature difference between the gases and the charge is as great as possible at all points (see Chapter 4). On the other hand, such a short flame and such intense combustion is frequently undesirable both because of the very severe effect on refractories which will result, and because the charge may require to be heated in a furnace having a large volume at a fairly uniform temperature. In many existing combustion appliances, however, the length of the flame is not fixed either by the requirements of optimum fuel economy or by those of uniformity, but is outside



the control of the operator. For example, in the case of pulverised fuel it is difficult to obtain complete burning of the particles without very large volume. It is therefore a very much needed step to improve the control of this very vital factor in combustion.

The mechanism by which flame length can be controlled has been the subject matter of the preceding four sections (3.2, 3.3, 3.4 and 3.5). Generally speaking, in furnace systems it is a matter of controlling the physical processes by which the two reacting materials are brought sufficiently close for the reaction to occur. Mechanisms for carrying out such control may involve the use of variable apertures by which the velocity and turbulence of one or other of the streams can be altered at will.

In the case of the control of flame length the observed variable should ideally be the distribution of temperature or radiation along the flame gases, although it is much more usual merely to observe the length of the luminous part of the flame. It is, however, possible to envisage a control system in which a turbulating mechanism is operated by the difference in temperature of two sensitive bodies or two heat-absorbing elements at different points along the flame. Such a mechanism could be made to give uniform temperature at these different points, or to give the compromise between the two which would be found to give good thermal efficiency without excessive wear on the refractories.

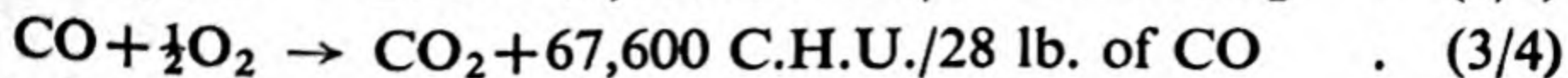
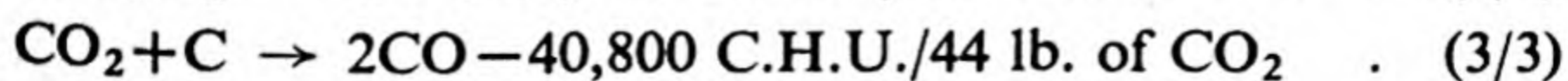
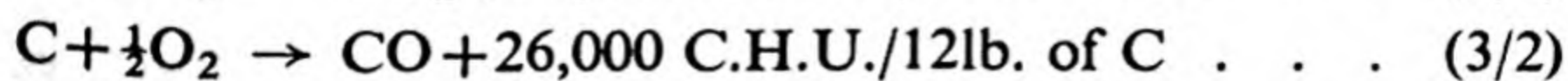
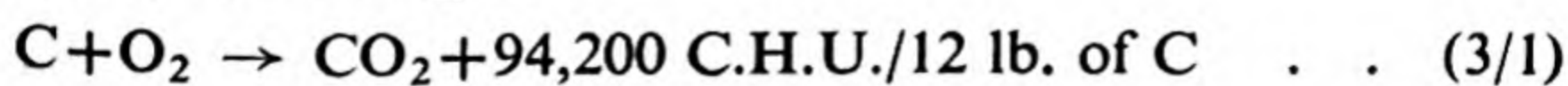
## SUMMARY OF PRACTICAL DATA IN CHAPTER 3

### 3.1. The Energy Release and Equilibria of Chemical Reactions

**3.1.1. Introduction.** Our knowledge of combustion should give answers to the questions :

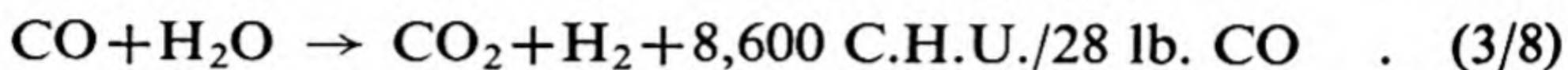
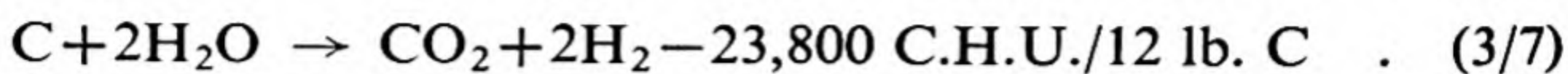
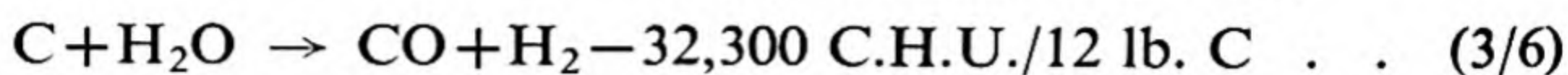
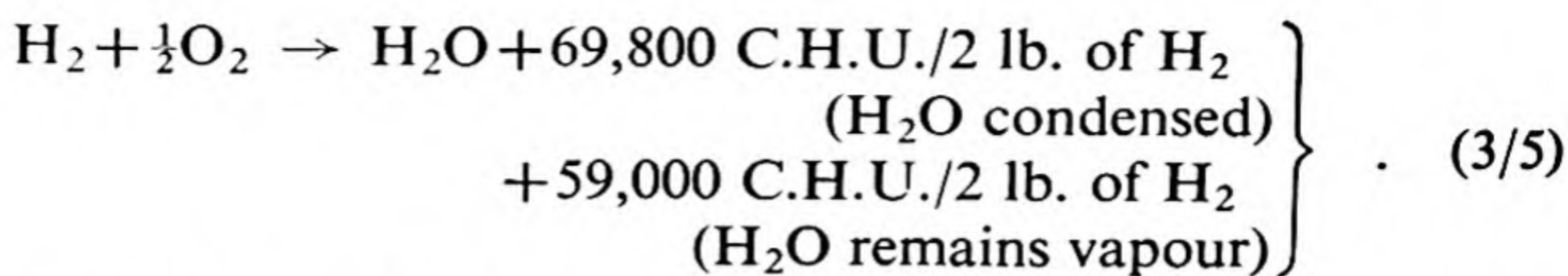
- (1) What is the energy released by completed reactions ?
- (2) What are the reaction equilibria ?
- (3) What are the reaction rates ?

#### 3.1.2. Reaction Energies





### 3.1.2 THE SCIENCE OF FLAMES AND FURNACES



**Theoretical volume of air/lb. of fuel :**

$$A_o = 1710 \left( \frac{C}{12} + \frac{H_2}{4} - \frac{O_2}{32} + \frac{S}{32} \right) n \text{ ft.}^3/\text{lb.} \quad (3/9)$$

Theoretical volume of combustion gases/lb. of fuel :

$$V_O = 359 \left\{ 4.76 \left( \frac{C}{12} + \frac{S}{32} \right) + 5.76 \frac{H_2}{4} + \frac{H_2O}{18} - 3.76 \frac{O_2}{32} + \frac{N_2}{28} \right\} n \text{ ft.}^3/\text{lb.} \quad (3/10)$$

$$\text{CO}_2 \text{ in wet combustion gases} = 300 \frac{C}{V_o} \quad (3/11)$$

$$\text{H}_2\text{O in wet combustion gases} = \frac{359}{V_o} \left( \frac{\text{H}_2}{2} + \frac{\text{H}_2\text{O}}{18} \right) \quad (3/12)$$

Theoretical volume of combustion gases (dry)/lb. of fuel ( $V'_o$ )

$$= 359 \left\{ 4.76 \left( \frac{C}{12} + \frac{S}{32} \right) + 3.76 \frac{H_2}{4} - 3.76 \frac{O_2}{32} + \frac{N_2}{28} \right\} \quad (3/13)$$

### Percentage CO<sub>2</sub> in theoretical dry combustion gases

$$= 300 \frac{C}{V'_{\rho}} (\text{CO}'_2) \quad . \quad . \quad . \quad . \quad (3/14)$$

Ratio of excess air to theoretical air (if percentage  $\text{CO}_2$  in actual dry combustion gases is found to be  $\text{CO}_2$ ) :

$$n = \frac{A - A_o}{A_o} = \frac{V'_o}{A_o} \left( \frac{CO'_2}{CO_2} - 1 \right) \quad . \quad . \quad . \quad (3/15)$$

( $A$  = actual air supplied).

If percentage  $O_2$  in actual dry combustion gases is found to be  $O_2$ ,

$$n = \frac{V'_o}{A_o} \frac{O_2}{0.21 - O_2} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (3/16)$$



Theoretical volume of air :

$$A_o = 4.76(3.67C_nH_m + 0.5CO + 2CH_y + 0.5H_2)n \text{ ft.}^3/n \text{ ft.}^3 \quad (3/17)$$

Theoretical volume of wet combustion gas

$$V_o = \alpha + \beta + \gamma \text{ ft.}^3/n \text{ ft.}^3 \quad . \quad . \quad . \quad (3/18)$$

where  $\alpha = CO_2 + 2.45C_nH_m + CO + CH_y$  is the volume of  $CO_2$   $n \text{ ft.}^3/n \text{ ft.}^3$  in the combustion gases ;

$\beta = 2.45C_nH_m + 2CH_y + H_2 + H_2O$  is the volume of  $H_2O$  in the combustion gases ; and

$\gamma = N_2 + \frac{3.76}{4.76}A_o$  is the volume of  $N_2$  in the combustion gases,

the theoretical volume of dry combustion gas  $V'_o = \alpha + \gamma$ .

### 3.1.3. Reaction Equilibria

For the reaction  $nA + mB = pC + qD$  the equilibrium constant  $K$

is given by 
$$\frac{1}{K} = \frac{(A)^n(B)^m}{(C)^p(D)^q} \quad . \quad . \quad . \quad . \quad . \quad (3/20)$$

and the variation of  $K$  with reaction temperature is given by the equation

$$-RT \log_e K = \Delta H - T\Delta S \quad . \quad . \quad . \quad (3/22)$$

When  $\Delta H$  (the heat of reaction at constant pressure) and  $\Delta S$  are independent of temperature this may be integrated in the form

$$K = A \cdot e^{-\Delta H/RT}$$

## 3.2. The Speeds of Reaction of Mixed Gases

**3.2.1. The Rates of Gaseous Reactions.** No general rules can be given for the reaction rates of perfectly mixed gases although the Arrhenius Law,  $k = A \cdot e^{-q/RT}$ , where  $q$  is an activation energy, can frequently be used. This is not applicable to chain reactions such as occur during the ignition of  $H_2$ ,  $CO$  and hydrocarbons.

**3.2.2. Flame Propagation Through a Gaseous Mixture.** The limits of inflammability of  $H_2$  in air at room temperature and pressure are 4 and 74%  $H_2$ , those for  $CO$  are 12.5 and 74%, and for  $CH_4$ , 5 and 15%.

**3.2.3. Combustion at a Burner Mouth.** The burning velocity of gas in Bunsen flame is  $U \sin \alpha$ , where  $U$  is the mean velocity in pipe and  $\alpha$  is the angle of the cone at a distance  $0.707R$  from axis. The mechanism of the burning in a steady flame is too complex for calculation of  $U$  on theoretical grounds.



**3.2.4. Surface Combustion.** For surface combustion the jet velocity must be very much greater than the burning velocity.

### 3.3. The Speed of Combustion in Diffusion Flames

**3.3.1. Examples of Diffusion Flames.** In large-scale flames where the gas and air are introduced side by side, mixing takes place by "eddy diffusion," which is much more rapid than the molecular diffusion which is the only mixing method in streamline flames.

**3.3.2. Streamline Diffusion Flames.** The rate-determining factor in a streamline flame is the diffusion, and as a result preheat has very little effect on the flame length.

**3.3.3. Turbulent Diffusion Flames.** Similarly, in large-scale flames the rate of combustion is entirely determined by the mixing rate, and again preheat has very little effect on flame length. The mixing rate can be greatly increased by having widely different air and gas velocities, narrow streams at a sharp angle to each other and by suitable brickwork on which the flame impinges.

An empirical formula for combustion in diffusion flames is

$$i = i_0(1 - e^{-kt}) \quad . \quad . \quad . \quad . \quad . \quad (3/26)$$

where  $k$  is a burner coefficient lying between 1.5 and 10,  $t$  is the time from leaving the burner mouth and  $i_0$  is the heat released by complete combustion. The volume necessary for a given fractional heat release is

$$\frac{Q}{k} \propto \text{ft.}^3 \quad . \quad . \quad . \quad . \quad . \quad (3/27)$$

where the volume coefficient  $\alpha$  varies between 13 and 176 (see Table 3/2). The most satisfactory method, however, for predicting combustion rates for preheated air flames is by studying the mixing rate with a model as discussed in Section 5.7.4.

**3.3.4. Practical Methods of Studying the Combustion Conditions in Diffusion Flames.** A convenient mixing index for gases leaving a furnace is given by

$$\frac{1}{2} \left( \frac{q_1}{q_2} + \frac{p_2}{p_1} \right)$$

where  $q_1/q_2$  and  $p_1/p_2$  are given from the ratio  $C_1/C_2$  of the percentage  $\text{CO}_2$  in the upper and lower halves of the channel and the ratio  $x/y$  of  $\text{CO}_2$  to  $\text{CO}$  in the original producer gas by

$$\frac{q_1}{q_2} = \frac{c_1}{c_2}; \quad \frac{p_1}{p_2} = \frac{(21 - c_1/2) + (21 - c_1)x/y}{(21 - c_2/2) - (21 - c_2)x/y} \quad . \quad . \quad (3/28)$$



### 3.4. The Speed of Combustion of Solid Fuels

**3.4.1. General Description of the Processes in Fuel Beds.** In a fuel bed it appears at first sight as if  $\text{CO}_2$  is at first formed and then reduced to CO.

**3.4.2. The Relative Carbon Saturation as a Function of Distance Through the Fuel Bed.** The course of the gas-carbon reactions can be simply studied using a single variable, the R.C.S. defined for pure carbon as

$$\frac{1 - 0.019\text{CO}_2 - 0.048\text{O}_2}{1 + 0.010(\text{CO}_2 + \text{O}_2)} \quad . \quad . \quad . \quad (3/29)$$

the R.C.S. is given by the relation

$$\text{R.C.S.} = A(1 - e^{-\alpha L/D}) \quad . \quad . \quad . \quad (3/30)$$

where  $A$  varies from 0.70 for beds with high heat loss to 0.97 for beds with low heat loss and air preheat and  $\alpha$  varies from 0.3 to 1.0.

**3.4.3. The Mechanism of Oxygen Consumption in the Early Part of the Fuel Bed.** In fact, the  $\text{O}_2$  is consumed in a fuel-bed by the production of CO which burns in the voids to produce the  $\text{CO}_2$  found in the early part of the bed.

**3.4.4. The Rate of Reduction of  $\text{CO}_2$  in Fuel Beds.** The  $\text{CO}_2$  reduction reaction is more complete at higher ratings because of reduced wall losses and is affected by reactivity.

**3.4.5. The Ignition of Fuel Beds.** Ignition is the state when the rate of heat liberation exceeds that of heat dissipation.

**3.4.6. Water-gas Reactions.** Where a hydrogen to  $\text{CO}_2$  ratio is required a reactive fuel and a low bed temperature are desirable; whereas when calorific value in water-gas is the primary object, high temperatures are desirable.

**3.4.7. Pulverised Fuel Firing.** The reaction rate per unit surface area of a pulverised fuel particle is given by

$$K = \frac{0.0109 \cdot 10^{-3} p_{\text{O}_2} T_p^{0.5}}{d} \text{ gm./sec. cm.}^2 \quad . \quad (3/32)$$

where  $p_{\text{O}_2}$  = partial pressure of  $\text{O}_2$  in atm.,

$T_p$  = particle temperature ( $^{\circ}$  K.),

$d$  = particle diameter (cm.).

### 3.5. The Speed of Combustion of Liquid Fuels

**3.5.1. Oil-firing Methods.** For furnaces oil is fired by breaking it up into a fine spray of droplets rather than by direct vaporisation.



### 3.5.2 THE SCIENCE OF FLAMES AND FURNACES

**3.5.2. Methods of “Atomising.”** The most usual method of making the spray of droplets is by using a high-velocity jet of air or steam in a special burner ; of these air is theoretically sounder, as it does not dilute the flame.

**3.5.3. The Factors Governing the Combustion of Atomised Oil.** Oil droplets burn by evaporation mainly caused by radiation falling on the droplet, followed by gas phase diffusion combustion of the vapour around each one.

**3.5.4. Air-fuel Mixing in Oil Combustion.** Injector burners entrain the air for combustion by virtue of the momentum of the fuel jet: proper burners fix the mixing pattern independently of the furnace chamber.

### 3.6. The Control of Combustion

**3.6.1. The Three Aspects of Control.** Ideally it should be possible to control separately the total rate of energy release, the quality of the energy release and the flame length.

**3.6.2. Control of the Quantity of Combustion.** The object of controlling the rate of energy release is to obtain the desired rate of heating of the charge.

**3.6.3. Control of the Quality of Combustion.** The object of combustion quality control is to use the minimum amount of fuel.

**3.6.4. Control of the Course of Combustion.** The object of flame-length control is to obtain the optimum temperature difference for heat transfer consistent with reasonable maintenance.



## 4. HEAT TRANSFER

### 4.1. Heat Transfer as a Thermodynamic Problem

**4.1.1. The Significance of the Heat Transfer Coefficient.** Having carried out the first of the energy conversion processes in a fuel-fired furnace, the conversion by combustion of chemical energy of fuel into thermal energy of flame gases, the furnace designer must next consider the second conversion process, the transfer of flame thermal energy to thermal energy of the charge to be heated. As before, his primary aims in controlling the second process are to achieve high output rates and low fuel consumption, and the first of these is attained by maximising the heat transfer rate per unit area of heat exchange surface, while the second is attained by carrying the gases out of the furnace with a temperature which is as little in excess of the lowest necessary charge temperature as possible. These conceptions can be made more precise in the following way. In general the rate of heat transfer ( $H$ ) between the combustion gases and the charge in a furnace is given approximately by a formula of the form

$$H = K_h \cdot A \cdot (\Delta T)^n$$

where  $K_h$  is a heat transfer coefficient whose characteristics will be discussed in the section 4.2 ;

$A$  is the area of the charge exposed to the flame ;

$\Delta T$  is the difference  $T_1 - T_2$  in temperature between the flame ( $T_1$ ) and the charge ( $T_2$ ) ;

$n$  is in most cases unity and in all cases near to unity for not too large values of  $\Delta T$ , e.g. when

$$\frac{\Delta T}{T_2} \gg \frac{1}{10}.$$

Suppose, therefore, it is required to heat a given quantity of material per hour in a furnace of given size.  $A$  and  $H$  are fixed : hence the product  $K_h(\Delta T)^n$  must also be fixed. Moreover, the temperature  $T_2$  of the charge is fixed by the heating requirements. Now the fraction of the heat of the gases which is used in the furnace depends primarily on the difference between the theoretical flame temperature  $T_0$  and the actual temperature  $T_1$  of the gases leaving the furnace ; hence to obtain good thermal efficiency  $T_1$  must be as



#### 4.1.1 THE SCIENCE OF FLAMES AND FURNACES

low as possible, i.e. as near to  $T_2$  as possible. Hence  $\Delta T$  must be a minimum while, as was shown above,  $K_h(\Delta T)^n$  is fixed. This means that a primary condition for good thermal efficiency is that  $K_h$  is a maximum. In other words, the whole problem of heat utilisation in a furnace resolves itself primarily into the problem of obtaining a high heat transfer coefficient in order to carry the minimum amount of virtue out of the furnace in the form of sensible heat of the heating gases. The study of heat transfer coefficients is thus a study of the way in which the requirements of thermodynamics can be satisfied by the most practical methods. Such a study is briefly summarised in 4.2, but before making it, it is necessary to give some consideration to the meaning of temperature in furnace flames and the methods of measuring them, since, by its definition, the determination of a heat transfer coefficient depends on knowing the temperature of the hot source.

4.1.2. **The Meaning of Temperature.** Thermodynamics is based on the study of systems in equilibrium. In a furnace there is no equilibrium of this type, since heat is continually being supplied, released and flowing away. In the steady state, however, there is a condition which is analogous to thermo-dynamic equilibrium in one respect, namely, in the fact that there are no changes *in time*, although it is quite dissimilar in that there are changes *in space*. Systems of this type have been called transflux equilibrium,<sup>4.1</sup> and it has been shown that where the energy flow is sufficiently violent the ordinary meaning of temperature may break down altogether. This is very rarely the case in furnace systems,\* but there is nevertheless one respect in which care is necessary in speaking of the temperature at any point of the furnace system. This arises from the fact that furnace gases are frequently fairly transparent to radiation. It follows that any given point in such gases there is a gas temperature and a radiant† temperature which may differ quite

\* Lewis and von Elbe<sup>4.2</sup> show that experimental flame temperatures significantly higher than the theoretical temperature (due to combustion being comparable in speed with the setting up of equilibrium between the different forms of energy) occur only in fast flames such as hydrogen and coal gas, and this phenomenon is not likely to have appreciable significance in furnace practice.

† The radiant temperature is defined as the temperature which would be taken up by a small black body at the point concerned if this body were completely unaffected by convection.

4.1 BENNETT, J. G., and PIRANI, M., "Symposium on Gas Temperature Measurement," *Inst. of Fuel* (1938).

4.2 LEWIS and VON ELBE: "Joint Investigation: Flame Temperature," *J. Appl. Phys.*, 11, p. 698 (1940).



considerably when, for example, the walls are much cooler than the flame. It will also frequently happen that the spectral distribution of the radiation does not correspond to that of a "black body" at any given temperature. It follows that the measuring of temperature in a flame is not a simple matter of inserting a thermocouple and reading the e.m.f. on a millivoltmeter.

✓ **4.1.3. Gas Temperature Measurement.** In considering the errors which occur in the measurement of gas temperature in furnace systems, it is necessary first of all to distinguish between, on the one hand, the errors due to the instrument used, e.g. conduction along the leads of a thermocouple, time lag in response when measuring fluctuating temperatures, and on the other hand, errors inherent in the principle of measurement. It is the latter type of error with which we are concerned. As discussed above, the gas at any given point in a furnace flame can be regarded as having a definite temperature, in other words, the various forms of energy in the gas are in equilibrium. The errors with which it is necessary to deal, therefore, are concerned with variation of temperature from point to point in the gas system and between the gas and the walls.

Where the gas temperature varies very much from point to point (e.g. along a vertical axis in a diffusion flame where the air is above the fuel gas) it is necessary to use different methods according to the use to be made of the measurement ; thus for the purpose of striking a heat balance the temperature of the gas at each height must be separately measured and an average calculated by weighting each such temperature according to the quantity flowing at that height, or alternatively the gas must be mixed in some way and the temperature of the mixture measured. Where radiation is concerned, however, these methods are quite unsuitable and temperature measurement has to be replaced by direct measurement of radiation giving the mean radiant temperature.

In many cases, however, it is sufficient to assume that the gas temperature is constant across the cross-section of the furnace, but differs appreciably from that of the walls. An example is the measurement of the temperature of heating and heated gases in a regenerator ; these are respectively much hotter than the walls and much colder. In this case the difficulty of temperature measurement arises from the fact that whether it is measured by a thermocouple, an optical pyrometer or a total radiation pyrometer, it will give a reading intermediate between the gas temperature and the wall temperature, being in all cases nearer to the gas temperature as



### 4.1.3 THE SCIENCE OF FLAMES AND FURNACES

the emissivity of the gas rises. There are six main ways of overcoming this difficulty and obtaining a fairly reliable measurement of the gas temperature :

- (i) The suction pyrometer.
- (ii) The two-colour pyrometer.
- (iii) The sensible-heat meter.
- (iv) The sodium-line reversal method.
- (v) The Kurlbaum method.
- (vi) Total-radiation comparative method.

Satisfactory ways of applying the last three methods to furnaces have not yet been worked out and they remain essentially methods for measuring the temperatures of flames in the laboratory. Nevertheless, it seems worth while including them as there is a chance that they could be modified so as to be of practical use in furnaces.

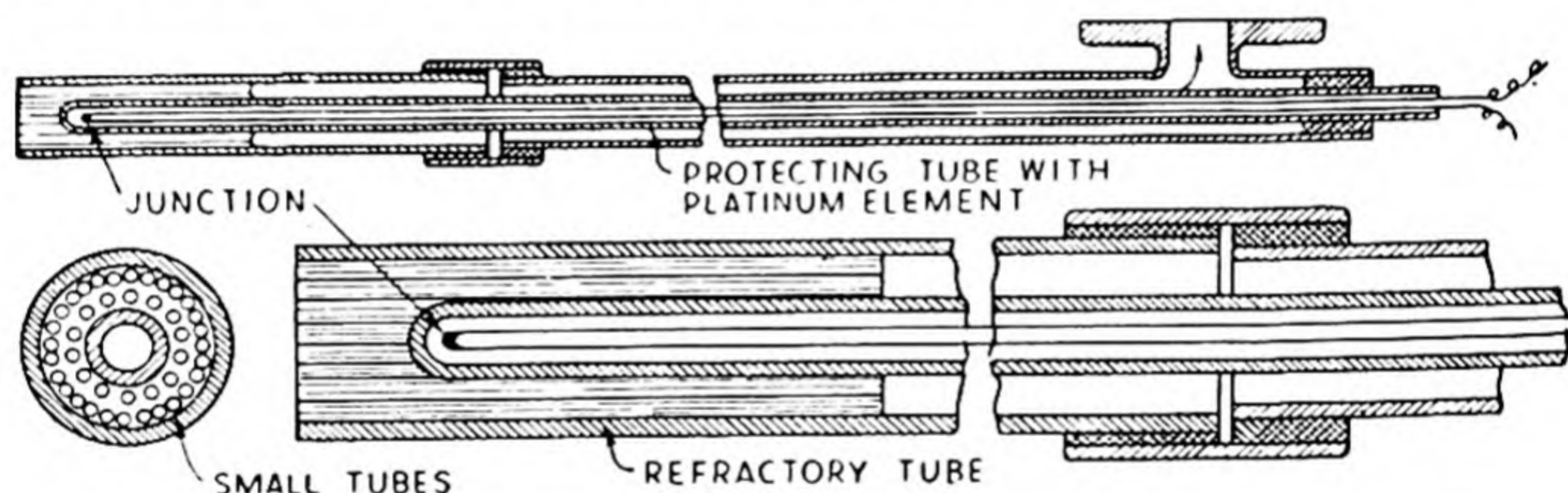


Fig. (4)1. Schack-type Suction Pyrometer with protecting ring of tubes.  
(Courtesy of the Institute of Fuel.)

(i) *The Suction Pyrometer*.<sup>4.3</sup> This consists, Fig. (4)1, of a thermocouple situated near the open mouth of a refractory tube, this mouth being placed near the point where the gas temperature is desired. The gas is sucked out through the tube at various rates, passing the tip of the junction, and the thermocouple reading is observed as a function of the gas suction rate. The suction rate is increased until further increase produces no further change in the temperature observed, and it is then assumed that forced convection on the tip of the couple has completely swamped the effect of the radiation from the walls of the furnace so that the reading corresponds to the true gas temperature. One modification of this type of instrument uses a venturi-shaped restriction around the thermocouple tip to produce maximum gas velocity at this point for a

<sup>4.3</sup> SCHACK, A. : "The Theory and Application of the Suction Pyrometer," *J. Inst. Fuel.*, 12, s. 31 (Mar. 1939).



given suction. Such instruments can have a heat-resisting metal arm or a silica arm for use at temperatures up to  $1100^{\circ}\text{C.}$ , and the thermocouple can be either bare or protected with a refractory. Platinum/platinum-rhodium thermocouples in high-grade mullite sheaths can be used up to  $1500^{\circ}\text{C.}$  with a water-cooled arm and special refractory tip,<sup>4.4</sup> but the method can only be used under research conditions and not for routine practice because of carbon particles and dust from the furnace, and because the couple is very easily contaminated by hydrogen or carbonaceous gases if the sheath becomes permeable. To overcome this difficulty proposals have been made to use suction pyrometers in which the gases are drawn through a refractory block with holes in it and an optical pyrometer is sighted on the inside of this block. This method is, however, subject to considerable difficulties both in the fact that the angle subtended by the block at the point where the optical pyrometer can be situated is very small, and again because of blocking up all the holes by dust and carbon particles. The development of ultra-sensitive total radiation and photocell units has made it possible to use a fine sheath with such an instrument located inside the water-cooled housing near it, and this type of instrument has a considerable future.

Greater accuracy has been achieved with suction pyrometers by surrounding the thermocouple sheath with a double circle of fine refractory tubes ("Schack tubes") so that convection heat transfer is increased and the effects of radiation heat transfer between the outside of the outer suction tube and the furnace walls are reduced. A still further refinement, which is desirable when the highest precision and certainty of results must be achieved, is the use of electric heating to balance the effects of heat loss by radiation. The measurement is necessarily more complicated in this case, however, as it is necessary to use a differential thermocouple to decide when the electric heating has exactly the right value, in addition to the thermocouple which measures the actual gas temperature. Refinements of this kind become more and more difficult to apply as the gas temperature rises from  $500^{\circ}\text{C.}$  to  $1500^{\circ}\text{C.}$ , and even simple suction pyrometers have not yet been successfully applied above the latter temperature.

(ii) *The Two-colour Pyrometer.* Two designs of optical pyrometer based on the use of two filters giving transmission at different colours

<sup>4.4</sup> HALLIDAY, I. M. D. : "Preheat Temperatures of Gas and Air," *I.S.I.*, Special Report No. 37, p. 48 (1946).



### 4.1.3 THE SCIENCE OF FLAMES AND FURNACES

have been put forward with the object of giving the true temperature of non-black bodies such as luminous flames.

In the method of Hottel and Broughton<sup>4.5</sup> a conventional optical pyrometer is arranged so that the lamp filament can be matched against the flame by adjustment of a rheostat, first with a red filter in the eye-piece and then with a green one, and the corresponding lamp temperatures can be deduced in each case from the lamp current. The true flame temperature and the emissivity of the flame can be calculated from these two temperatures using nomograms which have been prepared by the authors. These nomograms are based on the results of experiments on various luminous flames, from which it was concluded that the transmissivity of such flames is a simple function of the wavelength of the light. This relationship badly needs checking on furnace flames, however, and in any case it is very difficult to use the method when the flame flickers. See also the discussion on luminous radiation at the end of Section 4.2.4.

In the method of Naeser<sup>4.6</sup> a bichromatic, wedge-shaped light filter which transmits only red and green is used ; the coefficients of extinction of the two colours in the wedge differ, so that as the thickness of the wedge is increased green is extinguished more completely than red. This wedge is placed in series with a grey wedge in the system of an optical pyrometer in such a way that the field is matched against that from a standard source both for intensity and for colour. The scale of the colour wedge can be made to correspond to grey bodies or to bodies with any other fixed deviation from blackness. So long as the instrument is used on bodies with the same deviation from blackness as the calibration body, the colour temperature will be correct and the intensity temperature will be the temperature of the equivalent black body. For black bodies the two readings are equal, so that in other cases the difference between them gives an idea of the emissivity of the flame. This instrument has been used to some extent on luminous flames and is claimed to be reliable with them. The future of the two-colour instruments depends on the fuller investigation of the wavelength energy distribution from actual industrial flames and on the elimination of the human element by the use of the photocell.

4.5 HOTTEL and BROUGHTON : "Determination of True Temperature and Total Radiation from Luminous Gas Flames," *Ind. Eng. Chem., Anal. Ed.*, p. 166 (1932).

4.6 NAESER, G. : "Symposium on Gas Temperature Measurement," *Inst. Fuel*, p. s.38 (1939).



It has recently been shown by Wolfhard and Parker<sup>4.7</sup> that very small particles in flames introduce optical anomalies, so that the colour temperatures may be considerably higher than the true temperatures by an amount which depends on the size of the particles as well as their optical properties. Since luminous flames contain carbon particles which are in that size range ( $0.01-0.2\mu$ ) which gives such variation in properties, any calibration of a colour pyrometer based on the assumption that the correction factor depends only on the properties of carbon is likely to be in error. It is probable that colour temperature measurements on luminous

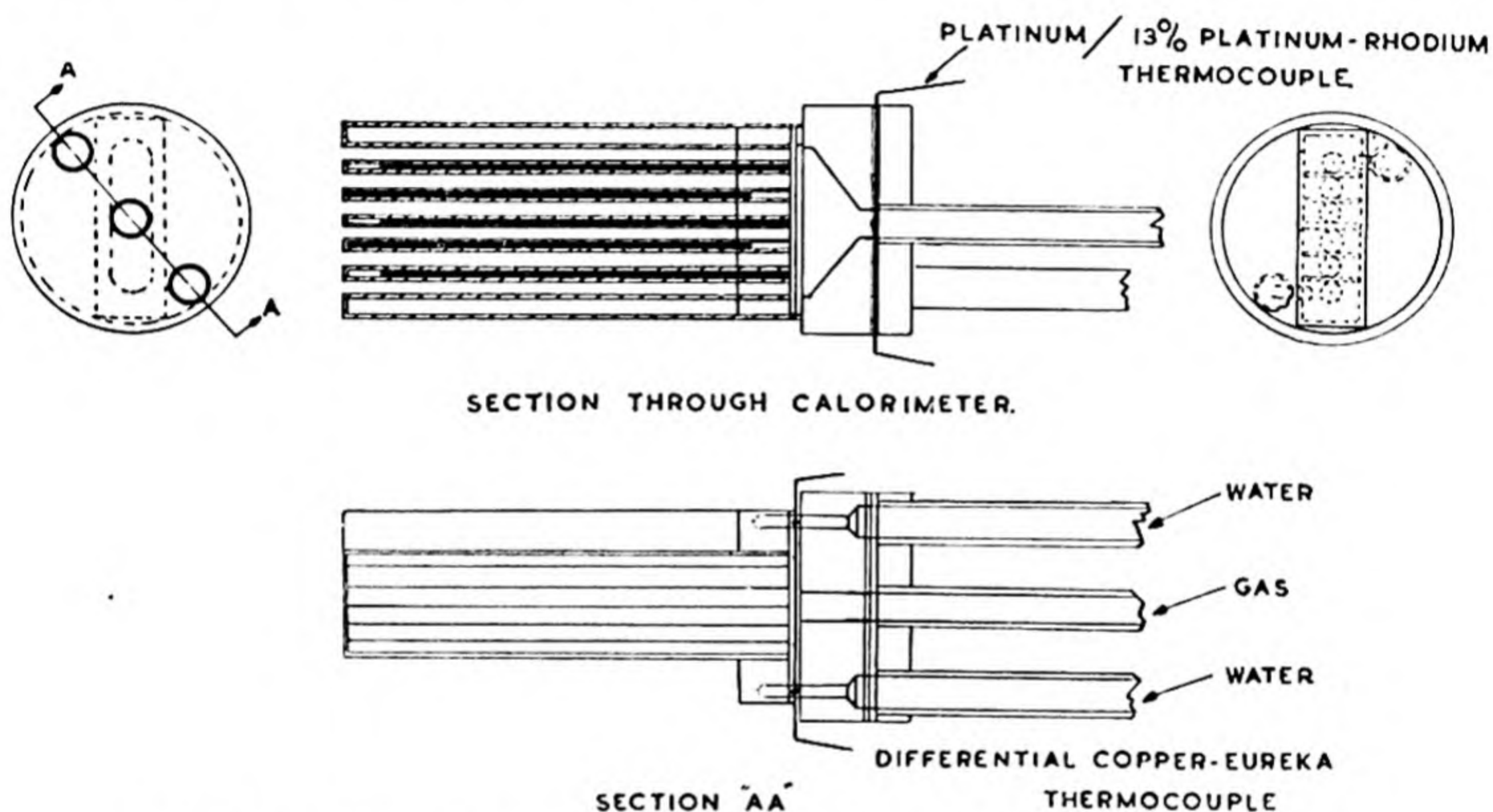


Fig. (4)2. Calorimeter for measuring the Sensible Heat of Flame Gases.

flames will only be of value if it happens that the size distribution of soot remains fairly constant in such flames and the instrument is calibrated empirically on a flame with the same size distribution.

(iii) *The Sensible Heat Meter.* The third method involves the use of a water-cooled calorimeter situated inside and insulated from an outer water-cooled protecting sleeve. When gas is drawn from the appropriate point through the calorimeter its sensible heat can be measured, while the quantity of gas sucked can also be measured after cooling. This measurement gives directly the sensible heat of the gas, the factor which is wanted for the heat balance (Section

4.7 WOLFARD, H. G., and PARKER, W. G.: "Temperature Measurement of Flames containing Incandescent Particles," *Proc. Phys. Soc. B.*, **62**, p. 523 (1 Aug. 1949).



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2.1.2) ; the true temperatures can be obtained from it from a knowledge of the specific heat of the gas. This instrument is shown in Fig. (4)2.

(iv) *The Sodium-line Reversal Method*.<sup>4.8</sup> This method is based on the assumption that the yellow radiation produced by sodium atoms introduced into a flame is in statistical equilibrium with the flame gases. This means that a sodium-coloured flame will obey Kirchhoff's law (see 4.2.4(iv)) at the sodium D wavelength just as at all other wavelengths, i.e. it will emit as much energy in this wavelength as it will absorb in this wavelength from a black body at the same temperature. The effect of the sodium is, however, to make the emissivity at the sodium D wavelength very much higher than that at neighbouring wavelengths. Hence if a radiator calibrated in terms of a black body at the sodium D wavelength, such as a tungsten filament lamp, is placed behind the flame and is viewed through the flame with a spectroscope, the sodium lines will appear brighter than the neighbouring spectrum if the filament is colder than the flame and vice versa. When the flame is hotter than the filament the flame adds more energy at the wavelength of the sodium line than it absorbs and the sodium lines stand out bright in the spectrum. When the temperatures are the same the sodium lines become invisible against the continuous spectrum. If, therefore, the tungsten lamp has been calibrated so that its radiation temperature is known as a function of the current put in, the temperature of the flame can be deduced from the reading of the current at which the sodium lines disappear. It will be seen that the necessary conditions for an easy match to be obtained are that the flame be fairly opaque at the wavelength of the sodium lines and a good transmitter at the neighbouring wavelengths. In neither case, however, do these conditions have to be perfectly satisfied for accuracy.

Another spectroscopic method of measuring flame temperatures is that of Coheur,<sup>4.9</sup> who photographed the rotational band spectrum of the flame and then determined the relative intensity of the various bands. The intensity of a band is proportional to the product of the number of molecules in the given vibrational state and the probability of the particular change of state ; the latter is known,

4.8 FERY, C. H.: *Comptes Rendus*, 137, p. 969 (1903).

4.9 COHEUR, P.: "Measurement of Temperature in Different Points of Flames and Arcs by the Spectroscopic Method," *Rev. Univ. des Mines*, 14, p. 650 (1938).



while the former can be directly calculated from the temperature if there is Boltzmann equilibrium, hence the temperature can be derived. In furnace flames such equilibrium will probably always exist, but in arcs or high intensity flames this is not necessarily the case. Coheur also gives a useful summary of other spectroscopic methods of measuring flame temperatures.

The difficulty of applying the sodium-line reversal method, like the next one, to furnaces, is the necessity to have a radiator on the other side of the flame from the observer and the presence of temperature gradients across the flame. Ribaud has overcome the latter defect by inserting the sodium in a narrow band into one part of a Bunsen flame, but industrial flames tend to contain sodium coloration at all points. Insertion instruments with an arrangement bearing a radiator spectroscopie and optical detector in a water-cooled housing which could be inserted into the furnace can, however, be envisaged and would overcome both defects.

(v) *The "Kurlbaum" Method*.<sup>4.10</sup> This method is only suitable for luminous flames. It is somewhat similar to the sodium-line reversal method except that as the luminous flame has the same range of wavelengths for emission as the black body, a different method of obtaining a match has to be used. A tungsten lamp is observed through the flame with an optical pyrometer and its temperature is adjusted until the reading of the pyrometer is the same whether the lamp is being observed with or without the flame interposed. This can be done by plotting the lamp current against the optical pyrometer reading without and then with the flame between, and the curve with the flame will cross the no-flame curve at the required point. Under these conditions the optical pyrometer reading is the true flame temperature, since once again the flame must be adding as much as it takes away from the light in the range observed. The flame must not be too transparent or else it becomes difficult to obtain any effect due to its insertion, nor must there be significant temperature differences across the luminous part of the flame or between the soot and the gas. Sakae Yagi<sup>4.11</sup> has found that the soot particles can be at a temperature widely different from that of the flame gases in a luminous flame, so that all temperature measuring methods are liable to be in error. Subject to these limitations, however, the method has been tried for furnace systems, but flicker

<sup>4.10</sup> KURLBAUM, F.: *Physik Zeits.*, 3, p. 187 (1902).

<sup>4.11</sup> SAKAE YAGI: "Studies on Luminous Flames, III," *Jl. Soc. Chem. Ind. of Japan*, 40, No. 4, p. 144 (1937).



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problems render the construction of the flame and lamp curve very difficult.

(vi) *The Total Radiation Comparative Method.* This method, which was proposed independently by Fishenden and Saunders and by Schmidt is based on the calculation from three total radiation measurements assuming that the flame is grey. The radiation from a flame alone ( $R_1$ ) is measured by using any form of small wall-type radiometer with a water-cooled target behind the flame. Similarly, the radiation from the flame and the wall ( $R_2$ ) is measured by the radiometer without the water-cooled target. The radiation from the wall alone ( $R_3$ ) can be measured by enclosing the radiometer in a suitable water-cooled housing, inserting it into the furnace across the flame and placing it close to the wall. From these readings the flame radiation temperature  $T_f$  and the flame emissivity may be calculated from the formulæ

$$\begin{aligned} R_1 &= \sigma E T_f^4 \\ R_2 &= \sigma E T_f^4 + (1 - E) R_3 \\ \text{whence } E &= \frac{R_3 + R_1 - R_2}{R_3} \\ \sigma T_f^4 &= \frac{R_1 R_3}{R_3 + R_1 - R_2} \\ T_f &= \sqrt[4]{\frac{R_1 R_3}{\sigma (R_3 + R_1 - R_2)}} \quad \dots \quad (4/1) \end{aligned}$$

where  $T_f$  = mean radiant temperature of gas layer concerned,  
 $E$  = emissivity of gas layer concerned.

This method has been used on small laboratory flames by Schmidt<sup>4.12</sup> and on large-scale flames by Sherman.<sup>4.13</sup> The accuracy of these formulæ clearly depends upon the assumptions (1) that  $E$ , the flame emissivity, is independent of wavelength and hence the absorption by the flame of radiation from the black body behind is the same fraction of the latter whatever the temperature of the black body ; (2) that the flame does not reflect.

The conditions of applicability of the method when the first of these conditions is not met may be examined in the following way. The equations for the simple method are set out in Table 4.1 in parallel columns with those in which this assumption is not met and the

<sup>4.12</sup> SCHMIDT : *Ann. der Phys.*, **29**, p. 998 (1909).

<sup>4.13</sup> SHERMAN : *A.S.M.E. Trans.*, **56**, p. 177 (1934) (see also 4.2.4 for an account of Sherman's results).



emissivity at a given wavelength  $e_\lambda$  is treated as a general function of the wavelength  $\lambda$ .

$R_1$  is the total radiation from the flame with a cold background ;

$R_2$  the total radiation from the flame with a hot background with temperature  $T_b$  ; and

$R_3$  the total radiation from the hot background alone, the latter being taken as a black body.

The function  $R_{\lambda T_b}$  is such that the energy emitted by a black body at temperature  $T_b$  °K. in the interval of wavelength  $\lambda$  to  $\lambda + d\lambda$  is  $R_{\lambda T_b} d\lambda$ .

$E$  is the overall emissivity of the flame in the case when the flame is assumed to be perfectly grey.

$E_b$  is the overall absorptivity of the flame for radiation from a black body at a temperature  $T_b$  passing orthogonally through it.

So

$$E_p = \frac{\int_0^\infty e_\lambda R_{\lambda T_b} \cdot d\lambda}{\int_0^\infty R_{\lambda T_b} d\lambda}$$

In the simplified treatment where the emissivity is assumed to be independent of temperature and hence of wavelength, it is calculated by the expression (1). The corresponding row on the right-hand side shows that this expression is the mean emissivity which the flame would have if it were at the temperature of the background hot body, but with the spectral emissivity-wavelength function unaffected. It follows from this that if the hot body behind it is at the same temperature as the flame, this expression gives the correct value for the wavelength-mean flame emissivity. Similarly, the expression (2), which on the left-hand side gives the total radiation from a black body at the flame temperature, gives on the right-hand side an expression which is the total radiation from a black body at the temperature of the background hot body multiplied by the ratio of the total radiation which the flame actually emits to that which it would emit if it were at the temperature of the hot body. Here again, therefore, if the hot body is at the same temperature as the flame, this expression, or  $R_3$  to which it reduces, is, as is obvious, the radiation from a black body at the flame temperature.

If, therefore,  $e_\lambda$  is a function of wavelength and not a constant,



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TABLE 4.1

A COMPARISON OF FORMULÆ FOR FLAME EMISSIVITY APPLICABLE WHEN THIS EMISSIVITY IS INDEPENDENT OF AND DEPENDENT ON WAVELENGTH.

Grey flame	Flame with emissivity varying with wavelength
$R_1 = \sigma E T_f^4$	$R_1 = \int_0^\infty e_\lambda R_{\lambda T_f} d\lambda$
$R_2 = \sigma E T_f^4 + \sigma(1-E)T_b^4$	$R_2 = \int_0^\infty e_\lambda R_{\lambda T_f} d\lambda + \int_0^\infty (1-e_\lambda) R_{\lambda T_b} d\lambda$
$R_3 = \sigma T_b^4$	$R_3 = \int_0^\infty R_{\lambda T_b} d\lambda = \sigma T_b^4$
$\therefore R_2 = R_1 + R_3 - E R_3$	$R_2 = R_1 + R_3 - \int_0^\infty e_\lambda R_{\lambda T_b} d\lambda$
$\therefore E = \frac{R_1 + R_3 - R_2}{R_3} \quad (1)$	$\frac{R_1 + R_3 - R_2}{R_3} = \frac{\int_0^\infty e_\lambda R_{\lambda T_b} d\lambda}{\int_0^\infty R_{\lambda T_b} d\lambda} = E_b \quad (1)$
$\sigma T_f^4 = \frac{R_1}{E} = \frac{R_1 R_3}{R_1 + R_3 - R_2} \quad (2)$	$\frac{R_1 R_3}{R_1 + R_3 - R_2} = \frac{\int_0^\infty R_{\lambda T_b} d\lambda \int_0^\infty e_\lambda R_{\lambda T_f} d\lambda}{\int_0^\infty e_\lambda R_{\lambda T_b} d\lambda} \quad (2)$

then the use of the expressions (1) and (2) to deduce the mean flame emissivity and the black body radiation temperature of the flame is in error.\* However, in the fundamental interpretation of the results for the deduction of mean flame emissivity and temperature three possibilities arise corresponding to the following cases :

- (1) When the flame is a close approximation to grey.
- (2) When the background is at a temperature not more than 100° C. from that of the flame.
- (3) When the background can be adjusted to two fixed temperatures bracketing the flame temperature or to any desired value to bring it to exactly the same temperature as the flame.

\* The use of the measurements  $R_1$ ,  $R_2$  and  $R_3$  to deduce the radiation heat exchange which the flame would have with walls (radiation  $R_4$ ) at a temperature equal to that of the background hot body by the formula

$$R_1 - R_4 \frac{R_1 + R_3 - R_2}{R_3} \quad \dots \quad (4/2)$$

remains exactly correct, since this expression (4/1) is equal to  $R_1 - E_b R_4$  and  $E_b = E_w$  in this case.



Dealing with these three possibilities in turn :

If the flame is nearly grey, then clearly the use of expressions (1) and (2) to deduce the mean flame emissivity and black-body radiation are justified, even though the background temperature is several hundred degrees different from that of the flame. It is quite likely that the deduction of  $E$  and  $\sigma T_f^4$  from expressions (1) and (2) will in fact be quite sufficiently accurate for the present purpose, since any method for flame-temperature measurement will be rendered liable to considerable error owing to the non-uniformity of temperature across the flame and other factors.

As far as the second method is concerned, a more detailed calculation has been carried out by H. Herne (private communication) for the case where the emissivity of the flame is assumed to be proportional to  $\lambda^{-0.9}$ , the function deduced by Hottel and Broughton from measurements on amyl-acetate flames. In order to eliminate regions in which the emissivity according to this formula exceeds unity the emissivity is taken to be zero for values of  $\lambda$  less than  $0.1/T$  cm., an assumption which introduces negligible error, since only a very small fraction of the black body emission at  $1600^\circ\text{C}$ . is thus neglected. This calculation shows that the error in taking the mean emissivity at the temperature of the background instead of at the temperature of the flame results in an error in this emissivity by the factor  $(T_f/T_b)^{0.9}$ . Thus, for example, if the furnace is  $100^\circ\text{C}$ . colder than the flame, the value of the emissivity deduced will be about 5% low. The value of the radiation from a black body of the same temperature as the flame, deduced from expression (2), will be in error by the reciprocal of the same fraction and hence it will be 5% high in the given case. The error in the value of  $T$  deduced from expression (2) will therefore be about a quarter of this, i.e. of the order of  $20^\circ\text{C}$ . for a flame temperature of  $1600^\circ\text{C}$ . When the correct temperature dependence of the flame emissivity is known, a curve can be plotted using this formula and the resulting expression will be even more accurate.

As regards the third alternative, this is really equivalent to an application of the Kurlbaum method to the total radiation of the flame instead of a narrow wavelength in the visible range only. Two ways of doing this are possible. In one the reading of the total radiation pyrometer is calibrated against the thermocouple in the black-body furnace to give a curve ; readings of the total radiation pyrometer are then taken for various values of the thermocouple, i.e. various furnace temperatures with the flame in between



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to give a second curve. When the furnace is colder than the flame, the readings of the total radiation pyrometer sighted through the flame will be higher than would correspond to the same thermocouple reading, and conversely. The point where the two curves cross, i.e. the thermocouple temperature at which  $R_2$  and  $R_3$  are equal, thus gives the true black-body flame temperature. The emissivity of the flame is then simply the ratio of  $R_1 : R_2$  and the method is free from error whatever the shape of spectral distribution curve.

This method requires, however, the alteration of the temperature of the hot background furnace. An alternative method is to have two hot background furnaces at two definite temperatures and to take readings of  $R_2$  at these two temperatures, the corresponding two values of  $R_3$  being known from previous calibration. By assuming that the curve for  $R_2$  is linear between these or has a normal degree of curvature, the crossover of the two curves can be deduced by interpolation and a method with high accuracy is obtained.

A method somewhat similar to that of Schmidt is to read the total radiation from various thicknesses of flame with a non-reflecting water-cooled target behind. The radiation for infinite thickness can be deduced by extrapolation according to the assumption that the emissivity of a thickness  $d$  is  $(1 - e^{-\alpha d})$ , where  $\alpha$  is unknown, and if the flame is luminous this infinite thickness radiation can be assumed equal to that of a black body at the flame temperature. This method has been used by Sherman, and by Mayorcas,<sup>4.14</sup> who found that the open-hearth furnace flame could be treated as luminous at the ingoing end and a reasonable temperature could be deduced by the method, but that at the outgoing end the radiation from the measured thickness was almost entirely non-luminous, so that the emissivity of an infinite thickness would only be about 20% of that of a black body at the same temperature.

## 4.2. The Laws of Heat Transfer

**4.2.1. The Mechanisms of Heat Transfer.** In this section only a very brief survey of the relevant physical concepts, laws and formulæ will be given, since the subject is fully and clearly covered in the *Calculation of Heat Transmission*, by Fishenden and Saunders, in *Heat Transmission*, by McAdams. Other useful texts are *Heat Transfer*, by M. Jakob and *An Introduction to Heat Transfer* by M. Fishenden and O. A. Saunders. The three well-known mechanisms by which heat is transferred (conduction, convection and

4.14 MAYORCAS, R.: *I.S.I. Special Report*, No. 37, p. 133.



radiation) are subject to quite different laws because the processes are quite different.

(i) In *conduction* the heat energy is handed on from molecule to molecule. Molecules with greater kinetic energy (temperature) impart some of their excess to those with less, by collision. In the case of solids which are electrical insulators, these collisions are between elastically bound atoms. In electrically conducting solids they are between the "free" electrons; and in gases (where, however, conduction plays a very small part unless the gas layers are only a few thousandths of an inch thick) the collisions are between the molecules.

(ii) In *convection* the heat is carried by actual physical movement of hotter materials to a colder region. The movement of the material may be in the form of independent molecules or in the form of eddies or pockets of fluid. Convection and conduction thus require the presence of matter, whether solid, liquid or gaseous.

(iii) *Radiation*, on the other hand, takes place most freely in the *absence* of matter. Heat radiation is similar to and obeys the same laws as the radiation of light, although it is usually studied rather from the point of view of a mechanism of energy transfer, whereas light is mostly used for the transfer of information.

**4.2.2. Conduction.** In conduction the quantity of heat  $dH$  which flows through a small area  $dA$  in a body in time  $dt$  is given by the basic equation:

$$dH = K dA \cdot dt \cdot \frac{dT}{dn} \quad (4/3)$$

where  $dT/dn$  is the temperature gradient in the direction normal to  $dA$ . The quantity  $K$  in this equation is known as the thermal conductivity of the object and is in general a function of temperature.

The value of  $K$  varies from  $3000 \frac{\text{C.H.U.}}{\text{ft}^2 \text{ hr. } ^\circ \text{C}} \left( 1 \frac{\text{cal.cm.}}{\text{cm}^2 \text{ sec. } ^\circ \text{C.}} \right)$  for copper to 0.2 (0.00007 c.g.s. units) for badly conducting solids. For liquids the variation is from 4 to 0.6 (0.0014 to 0.0002 c.g.s. units) and for gases from 1 to 0.05 (0.00035 to 0.000017 c.g.s. units). In furnace studies, however, thermal conductivity is only of importance in connection with solids,\* since its function as a

\* The thermal conductivity of gases does, however, play a role in deciding their convection behaviour and a knowledge of it is essential in order to be able to estimate the results to be expected with one gas in one set of conditions from those obtained with another gas in another set of conditions (see 4.2.3 below).



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mechanism of heat transfer is completely swamped by that of convection in the case of gases and usually also in the case of liquids. The only exceptions to this rule are the case of dense liquids such as molten metals, and salt baths.

When the temperature at various points in a conductor is being altered by the flow of heat the general differential equation governing the temperature  $T$  at a point  $xyz$  in a solid of thermal conductivity  $K$  can be derived from (4/3) to be

$$K\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) = C \frac{\partial T}{\partial t} \quad . \quad . \quad . \quad . \quad (4/4)$$

where  $C$  is the specific heat per unit volume.

Two cases of conduction should be distinguished: the *steady state*, where heat flows continuously from a source to a sink without increasing the temperature, and the *unsteady state* where the body through which heat is flowing is rising in temperature during the process. In the steady state the term on the right-hand side of equation (4/4) vanishes. In both steady and unsteady state the ultimate mathematical equations and all the physical data can often be exactly specified, but nevertheless the problem cannot be accurately solved merely due to the complexity of the shapes involved, because the differential equation (4/4) cannot be integrated except where the boundary conditions are simple. In the case of uniform parallel heat flow it becomes

$$KdT/dx = \frac{1}{A} dH/dt \quad . \quad . \quad . \quad . \quad (4/5)$$

In the case of steady-state heat flow, such as occurs in a continuous furnace after a considerable period of heating up, the complexity is limited to complexity of shape. Examples of such awkward geometrical conditions are the corners of a furnace or the hearth of a furnace resting on the ground.<sup>4.15</sup> In most cases, however, it is possible to obtain the desired accuracy ( $\pm 20\%$  is usually a great deal better than nothing) by means of intelligent simplification: for example, in calculating heat losses at the corner of a furnace it is quite accurate enough to use formula (4/5) and take for  $A$  the average area halfway between that of the inside and outside. It is on some occasions desirable to work out the exact isotherms, but in many cases the complexity of these calculations is such as not to justify making them, the use of the very simplest approximations being preferable, since in any case the physical magnitudes are rarely

<sup>4.15</sup> KELLAR, J. D.: *Trans. A.S.M.E., F.S.P.* 50-37, p. 111 (1928).



known with any accuracy. Graphical methods for the solution of steady-state heat flow in various shapes have been put forward by Awbery,<sup>4.16</sup> and a number of methods based on the similarity of the laws of heat flow with the laws of flow of electricity in electrolytes have been used in special cases.

Unsteady-state heat transfer has, in addition to geometrical complexity, the complication of time variation, but similarity methods can again be used, such as electrical networks<sup>4.17</sup> and the flow of liquids in tubes.<sup>4.18</sup> Calculations of unsteady-state heat transfer are of importance in furnace design in connection with the heating up of large metal blocks and the storage of heat in the walls of furnaces with fluctuating temperatures. This problem has had a good deal of attention in furnace design because, for example, the rate of heating of various parts of a large steel bar may decide whether the bar is useful or must be rejected. In the case where the surface material is instantly raised to a certain temperature and maintained there, the temperature distribution at any specified time has been calculated for different shapes of block. The problem is considerably more difficult in the more practical case where the bar is instantly subjected to radiation from surroundings at a definite temperature or a certain fixed rate of heat input is suddenly switched on. This matter is discussed further in 4.5 below.

**4.2.3. Convection.** Convection is very closely bound up with the laws of flow of fluids, and just as fluid flow shows a transition from streamline to turbulent as the flow rate is increased, so the convection process shows a transition. In the case of conduction we saw that in many applications the differential equations could not be solved owing to the mathematical complexity, although the physical processes were reliably represented by these equations. In the case of convection, however, it is not even possible to write down the differential equations governing the process; consequently, the

<sup>4.16</sup> AWBERY, J. H., and SCHOFIELD, F. H.: *Proc. Intern. Congr. Refrig.*, 5th Congr., 3, pp. 591-610 (1929).

<sup>4.17</sup> JACKSON, R., SARJANT, R. J., WAGSTAFF, J. B., EYRES, N. R., HARTREE, D. R., and INGHAM J.: "Variable Heat Flow in Steel," *J.I.S.I.*, 11, 211P (1944).

EYRES, N. R., HARTREE, D. R., INGRAM, J., JACKSON, R., SARJANT, R. J., and WAGSTAFF, J. B.: "The Calculation of Variable Heat Flow in Solids," *Phil. Trans. Roy. Soc., Series A*, 240, pp. 1-57 (2 Aug. 1946).

PASCHKIS, V., and BAKER, H. D.: "A Method for Determining Unsteady-state Heat Transfer by Means of an Electrical Analogy," *Trans. A.S.M.E.*, 64, p. 105 (1942).

<sup>4.18</sup> MOORE, A. D.: "The Hydrocal," *Ind. Eng. Chem.*, 128, p. 704 (1936).



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application of mathematical methods is considerably more limited than it was in the case of conduction. Recently, however, a great deal of progress has been made in this field by the use of the principle of similarity first introduced in connection with fluid flow by Osborn Reynolds<sup>4.19</sup> and first applied to convection by Rayleigh.<sup>4.20</sup> An excellent account of the principle of similarity is given in Appendix A of *Calculation of Heat Transmission*, by Fishenden and Saunders. The principle is an expression of the fact that a *physical* system cannot know the units in which *men* are measuring it, consequently the state of the system must be governed not by the lengths, velocities or temperature that men measure in their arbitrary units, but by ratios of these quantities which are free from dimensional units. When, therefore, the results of a series of experiments can be expressed in the form that one dimensionless criterion (A) is a certain numerical function of another dimensionless criterion, (B) ( $A=f(B)$ ), these results become applicable to any system in which the same physical process is going on provided the system  $Q$  has the same value of the criterion ( $b$ ) even though  $Q$  may consist of an entirely different fluid and boundaries ten times as far apart. ✓

What the principle of similarity cannot tell one, however, is which dimensionless criteria apply to any given problem. It is necessary to examine the system very carefully to make sure that one dimensionless criterion is inserted for every physical factor which enters into the problem. Thus the dimensionless criterion involving  $H$  (the heat transfer per unit area and time C.H.U./ft.<sup>2</sup>-hr. or cal./cm.<sup>2</sup> sec.) from a surface is  $HD/K\theta$ ,

where  $D$  is a characteristic length of the system (ft. or cm.),

$K$  is the thermal conductivity of the gas concerned  
(C.H.U.-ft./ft.<sup>2</sup>-hr. ° C. or cal./cm.<sup>2</sup> sec. ° C.),

$\theta$  is the temperature difference (° C.).

In general, for a convection heat transfer system, this quantity  $HD/K\theta$  involving the dependent variable  $H$ , which is known as the Nusselt number.  $N$ , is a function of two or more dimensionless groups involving the independent variables of gravity, momentum and viscosity. We shall see, however, that for practical purposes it

4.19 REYNOLDS, O.: *Phil. Trans. Roy. Soc.*, **174**, pp. 935-982 (1883): *Scientific Papers*, **2**, 51-105.

4.20 RAYLEIGH: *Phil. Mag.*, **59**, p. 34, 1892: **8**, p. 66, 1904. *Nature*, **95**, p. 66, March 1915.



is usually sufficient to express  $N$  as a function of a single dimensionless group and thus obtain a workable formula. The application of similarity to convection problems is essentially based on the assumption that the whole temperature drop  $\theta = T_1 - T_2$  takes place across a thin film of stagnant gas in which heat is transferred by conduction and the thickness of which depends on the rate of flow of gas. It is because this film occupies a very small part of the whole gas volume while the rest of the gas is at a substantially uniform temperature that it is possible to express all the gas properties in terms of the mean temperature of the gas. In choosing the dimensionless groups it is necessary to distinguish two types of convection, natural and forced convection.

(a) In the case of *natural convection* the gas flow which carries the heat from the surface is produced by the temperature difference itself, which causes a difference of density in different parts of the gas and hence a convection current. This type of heat transfer thus involves the coefficient of thermal expansion of the fluid ( $\alpha \text{ } ^\circ\text{C.}^{-1}$ ) and the acceleration due to gravity ( $g \text{ ft./hr.}^2$  or  $\text{cm./sec.}^2$ ).<sup>\*</sup> The dimensionless criterion involving these factors is the Grashof number

$$G = \frac{ag\theta D^3 C}{K\nu}$$

which may conveniently be written as  $\phi \cdot \theta \cdot D^3$ , where

$$\phi = \frac{agc}{K\nu} \cdot \frac{1}{\text{ft.}^3 \text{ } ^\circ\text{C.}}$$

$\phi$  being thus a function of the properties of the gas which can be read direct from the last column of Table 4.1, where  $c$  = specific heat of fluid/unit volume (C.H.U./ft.<sup>3</sup>  $^\circ\text{C.}$  or cal./cm.<sup>3</sup>  $^\circ\text{C.}$ ) at constant pressure. A second dimensionless criterion which governs the ratio of momentum transfer by viscosity to heat transfer by conduction is the quantity  $C\nu/k$ ,<sup>†</sup> where  $\nu$  equals the kinematic viscosity of gas. For all diatomic gases and for the triatomic  $\text{CO}_2$ , however, the quantity  $C\nu/k$  has very nearly the same value, viz. 0.74; for the triatomic gas  $\text{H}_2\text{O}$  it is about 0.8–0.9. Hence there is no need to take account of it in transferring results obtained with one gas such as air to a system with another similar gas. The following general formulæ have been calculated from the results of various workers for the convection loss from horizontal cylinders in any gas at any temperature and may be relied upon to  $\pm 15\%$ .

<sup>\*</sup>  $g = 4.17 \times 10^8 \text{ ft./hr.}^2$  or  $981 \text{ cm./sec.}^2$

<sup>†</sup> Known as the Prandtl number.



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For  $G = 10^3 - 10^9$ ,  $HD/K\theta = 0.47 \cdot G^{0.25}$  . . . . (4/6) (i)  
or  $H = 0.47K(\phi/D)^{0.25} \theta^{1.25}$  . . . . (ii)  
For  $G > 10^9$   $HD/K\theta = 0.10G^{0.33}$  . . . . (iii)  
or  $H = 0.10 \cdot K\phi^{0.33} \theta^{1.33}$  . . . . (iv)

Similar formulæ can be derived for the other main industrial surface arrangements. However, for hot surfaces exposed to air at normal temperatures these formulæ may be still further simplified.

TABLE 4.1  
PHYSICAL CONSTANTS FOR GASES AT DIFFERENT TEMPERATURES

Temperature, <i>t</i>	Thermal conductivity, <i>K</i>	Specific heat per unit volume at constant pressure, <i>c</i>	Kinematic viscosity, <i>ν</i>	$\phi = \frac{agc^*}{K\nu}$
° C.	C.H.U./ft. hr. ° C.	C.H.U./ft. <sup>3</sup> ° C.	ft. <sup>2</sup> /hr.	10 <sup>1</sup> /ft. <sup>3</sup> ° C.
Air, CO, O <sub>2</sub> , N <sub>2</sub>				
0	1.37 × 10 <sup>-2</sup>	1.94 × 10 <sup>-2</sup>	0.52	0.416
200	2.08 × 10 <sup>-2</sup>	1.15 × 10 <sup>-2</sup>	1.34	0.0364
400	2.73 × 10 <sup>-2</sup>	0.84 × 10 <sup>-2</sup>	2.42	0.00793
600	3.27 × 10 <sup>-2</sup>	0.69 × 10 <sup>-2</sup>	3.59	0.00265
800	3.86 × 10 <sup>-2</sup>	0.57 × 10 <sup>-2</sup>	5.15	0.00112
1000	4.34 × 10 <sup>-2</sup>	0.465 × 10 <sup>-2</sup>	6.7	0.000533
1200	4.76 × 10 <sup>-2</sup>	0.435 × 10 <sup>-2</sup>	8.4	0.000308
1400	5.09 × 10 <sup>-2</sup>	0.388 × 10 <sup>-2</sup>	10.1	0.000189
1600	5.3 × 10 <sup>-2</sup>	0.350 × 10 <sup>-2</sup>	11.7	0.000124
CO <sub>2</sub>				
0	0.84 × 10 <sup>-2</sup>	2.38 × 10 <sup>-2</sup>	0.276	1.564
200	1.72 × 10 <sup>-2</sup>	1.66 × 10 <sup>-2</sup>	0.78	0.110
400	2.58 × 10 <sup>-2</sup>	1.31 × 10 <sup>-2</sup>	1.46	0.0220
600	3.42 × 10 <sup>-2</sup>	1.09 × 10 <sup>-2</sup>	2.28	0.00661
800	4.13 × 10 <sup>-2</sup>	0.93 × 10 <sup>-2</sup>	3.25	0.00268
1000	4.74 × 10 <sup>-2</sup>	0.81 × 10 <sup>-2</sup>	4.31	0.00129
1200	5.2 × 10 <sup>-2</sup>	0.71 × 10 <sup>-2</sup>	5.47	0.000726
1400	5.7 × 10 <sup>-2</sup>	0.64 × 10 <sup>-2</sup>	6.65	0.000412
1600	6.1 × 10 <sup>-2</sup>	0.58 × 10 <sup>-2</sup>	7.7	0.000270
H <sub>2</sub>				
0	9.9 × 10 <sup>-2</sup>	1.91 × 10 <sup>-2</sup>	3.62	0.810
200	14.5 × 10 <sup>-2</sup>	1.12 × 10 <sup>-2</sup>	9.2	0.0740
400	18.3 × 10 <sup>-2</sup>	0.79 × 10 <sup>-2</sup>	16.3	0.0165
600	22.0 × 10 <sup>-2</sup>	0.62 × 10 <sup>-2</sup>	25.3	0.00527
800	25.7 × 10 <sup>-2</sup>	0.52 × 10 <sup>-2</sup>	35.8	0.00216
1000	29.2 × 10 <sup>-2</sup>	0.446 × 10 <sup>-2</sup>	47.0	0.00106
1200	32.2 × 10 <sup>-2</sup>	0.393 × 10 <sup>-2</sup>	59	0.000584
1400	34.9 × 10 <sup>-2</sup>	0.356 × 10 <sup>-2</sup>	70	0.000359
1600	37.4 × 10 <sup>-2</sup>	0.325 × 10 <sup>-2</sup>	84	0.000228

\* Adapted by kind permission from "Heat Transfer" Data, by Fishenden, M., and Saunders, O. A., p. 363 *et seq.* of *Waste Heat Recovery from Industrial Furnaces* (Chapman and Hall, 1948).



- (1) Vertical surface in air

$$H=0.35 \cdot \theta^{5/4} \text{ C.H.U./ft.}^2 \text{ hr.}$$

or  $H = 4.6 \cdot 10^{-5} \cdot \theta^{5/4}$  cal./cm.<sup>2</sup> sec. . . . . (4/7)

- (2) Horizontal surface facing upwards,  $H$ , calculated from 4/7, should be increased by 30%.

- (3) Horizontal surface facing downwards,  $H$ , calculated from 4/7, should be reduced by 35%.

- (4) Horizontal cylinder in air

$$H = 0.40 \theta^{5/4} \text{ C.H.U./ft.}^2 \text{ hr. } (H = 5.3 \cdot 10^{-5} \theta^{5/4} \text{ c.g.s. units}) \quad (4/8)$$

The reason that  $\theta$  (the temperature difference) appears to a power slightly above unity for natural convection is that it affects not only the amount of heat conducted across the stagnant layer, but also by virtue of the effect on the thermal expansion it affects the velocity of gas flow and hence the thickness of the stagnant layer.

(b) *Forced convection* applies to the case where the flow of the gas is induced not by the change in density due to the heat transfer but by some other cause, e.g. the general flow of gas through a furnace system caused by a fan or stack. In the region where these velocities are very low, natural convection will play a part in addition to forced convection. An estimate can be made as to whether the system is in this region by calculating the natural convection heat transfer, assuming there is no gas flow, by one of the formulæ (4/6)–(4/8) and calculating the forced convection heat transfer by the formulæ given below and thus seeing whether the latter swamps the former.

In the case of forced convection the most important dimensionless criterion is that containing the independent variable  $V$ , the velocity, viz.  $Vc.D/K$ .\* As in the case of natural convection, the second criterion,  $c\nu/K$  can be treated as constant, as it is approximately the same for all diatomic gases and the error due to neglecting it when there is the normal percentage of triatomic water vapour occurring in furnace gases is small in comparison with the general error of all such calculations. Since  $c\nu/K$  is constant, the Reynolds number  $VD/\nu$  obtained by dividing  $Vc.D/K$  by  $c\nu/K$  can

\* In this equation  $v$  and  $c$  refer to the velocity and specific heat per unit volume of the *hot* gas respectively. Since, however,  $Vc = V_0c_0$ , where the suffix  $_0$  refers to the N.T.P. condition, the criterion  $V_0c_0D/K$  can equally well be used.



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alternatively be used as the dimensionless criterion involving the velocity.

Each geometrical arrangement for forced convection has to be considered separately and a number of systems have been extensively studied. Of these,

- (1) the forced convection to a tube bundle is of importance in connection with air heaters ;
- (2) the forced convection to a single cylinder from a gas flowing outside it at right angles to the axis is of importance in connection with the heating of metal bars by furnace flames and is probably the most important case in furnace calculations ;
- (3) the forced convection between a tube and a gas flowing through it can be used for studying the heat transfer to the charge and furnace walls where the gases are flowing along large surfaces. In all these cases, however, it is necessary to allow for flame radiation which will swamp the convection at high temperatures except where the gas velocities are very high. The formula for tube bundles is (cf. Lander<sup>4.21</sup>)

$$H = 0.32 C_H \left( \frac{VD}{\nu} \right)^{0.6} \frac{K\theta}{D} \quad \dots \quad (4/9)$$

(C.H.U./ft.<sup>2</sup> hr. when in British units, cal./cm.<sup>2</sup> sec. in metric units)

where  $C_H$  is a factor which takes account of the particular tube arrangement used and varies between 0.66 and 1.05 for "straight-through" tube bundles and between 0.94 and 1.20 for staggered arrangements; the velocity in this formula refers to the velocity in the narrowest restriction.

A nomogram, Fig. (4)3a, is given to evaluate the expression

$$0.32 \frac{K}{D} \left( \frac{VD}{\nu} \right)^{0.6}$$

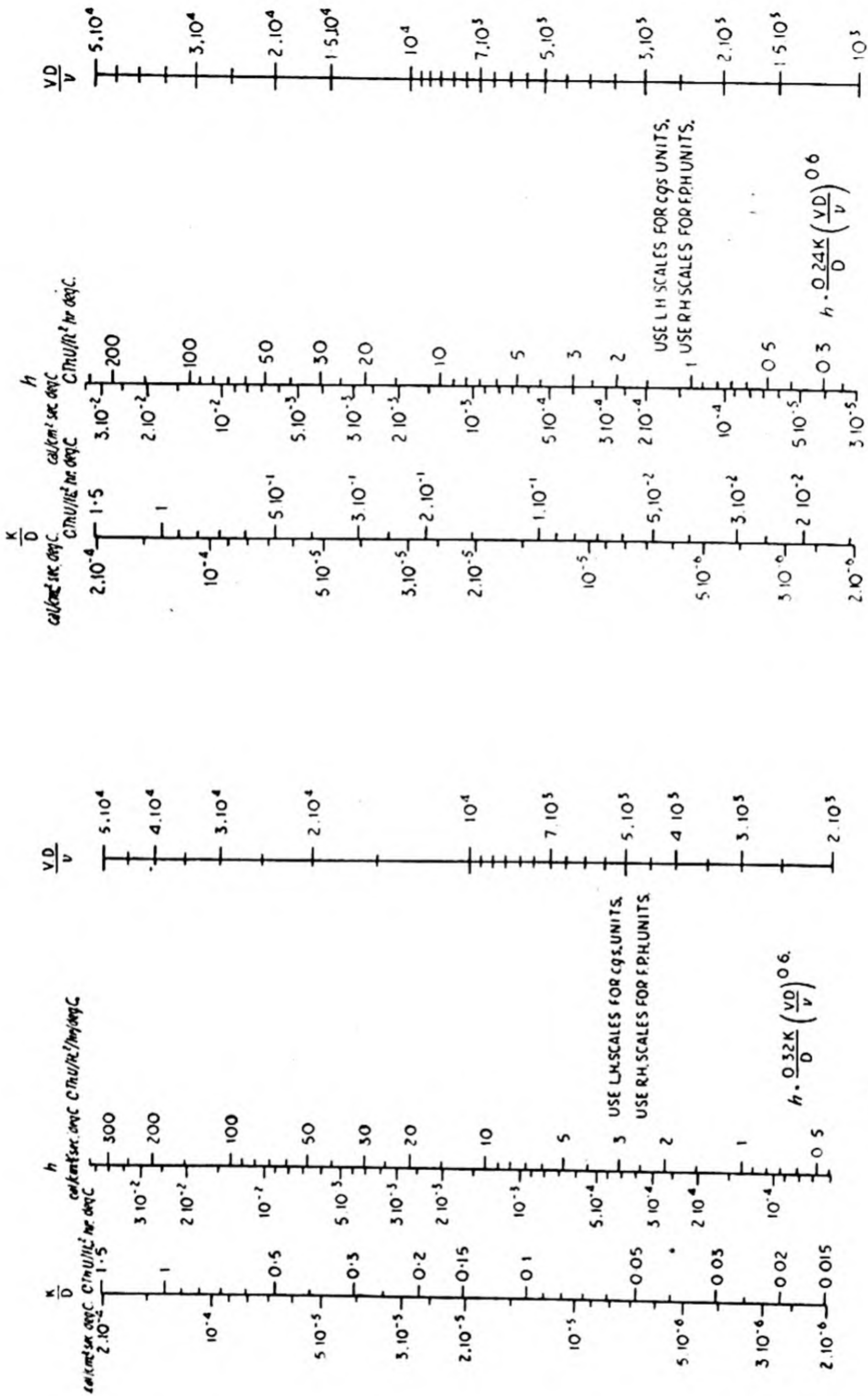
For a gas flowing at right angles across the outside of a cylinder the following formulæ can be used :

$$H = \left[ 0.32 + 0.43 \left( \frac{VD}{\nu} \right)^{0.52} \right] \frac{\theta K}{D} \text{ for } \frac{VD}{\nu} \text{ between } 0.1 \text{ and } 1000 \quad (4/10) \text{ (i)}$$

$$H = 0.24 \left( \frac{VD}{\nu} \right)^{0.6} \frac{\theta K}{D} \text{ for } \frac{VD}{\nu} \text{ between } 1000 \text{ and } 100,000 \quad (4/10) \text{ (ii)}$$

<sup>4.21</sup> LANDER, C. H. : "A Review of Recent Progress in Heat Transfer," *Proc. Inst. Mech. Eng.*, **148**, p. 81 (1942).





(a)

(b)

Fig. (4)3. (a) Nomogram for calculating forced convection between a tube bank and a gas flowing at right angles to the bank.  
(b) Nomogram for calculating forced convection with a gas flowing round a tube at right angles.



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In the case of convection between a tube and a gas flowing inside it, the formula is

$$H = 0.021 \left( \frac{VD}{\nu} \right)^{0.8} \frac{K\theta}{D} \quad \dots \quad (4/11)$$

In formulæ 4/9-4/11 the units of  $H$  are determined by those of  $K$ ,  $D$  and  $\theta$ ; since  $HD/K\theta$  is dimensionless, the coefficient is unchanged by a change of units. Also the Reynolds number group,  $VD/\nu$ , is the same in any self-consistent set of units. The

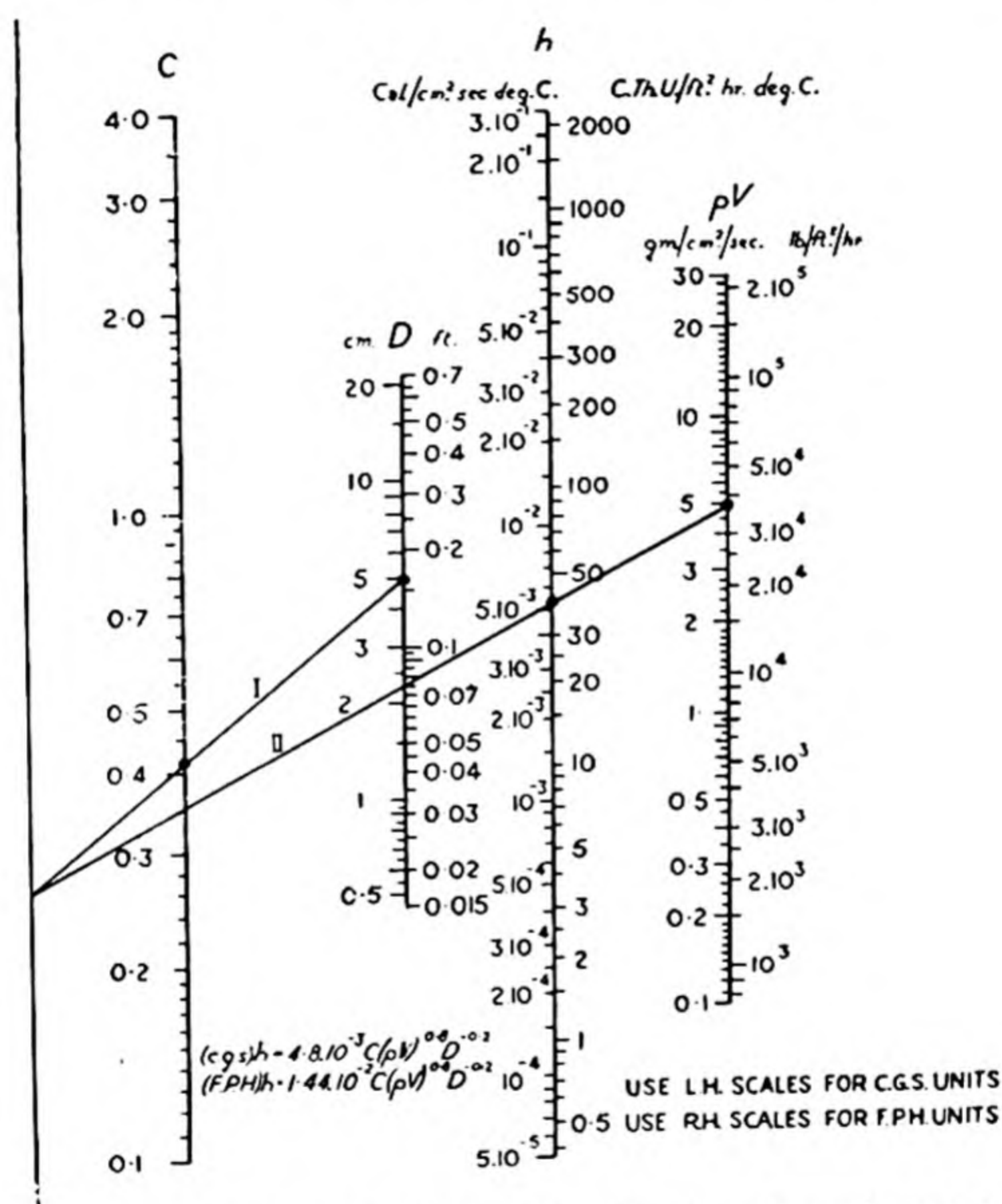


Fig. (4)4. Nomogram for calculating forced convection with a gas flowing inside a tube.

nomogram of Fig. (4)3b has been drawn to give a quick method of calculating forced convection for a gas on the outside of a cylinder, whilst the nomogram of Fig. (4)4 is for a gas inside a cylinder (eq. 4/11).

One other case of forced convection is of importance. This is the heat transfer between a stream of gas and a bed of broken solids.



Saunders and Ford<sup>4.22</sup> have shown that for a value of the dimensionless criterion

$$\frac{VDc}{K_s} \text{ below approximately } 6 \quad . \quad . \quad . \quad (4/12)$$

where  $K_s$  is the thermal conductivity of the solids,  $D$  their average diameter and  $c$  = specific heat of gas, the thermal conductivity of the solids can be regarded as infinite, i.e. the whole resistance to heat transfer is that which occurs in the stagnant film between the gas and the solid. When this condition is satisfied,  $H$ , the heat transfer coefficient from the gas to the solid per unit volume of the bed, is given by the equation

$$H_v = \frac{67V'(1-f)}{D} \frac{\text{C.H.U.}}{\text{ft.}^3 \text{ } ^\circ\text{C.}} \text{ or } \left( \frac{H=0.0003V'(1-f)}{D} \text{ cal./cm.}^3 \text{ sec.} \right) \quad (4/13)$$

The symbols have the following significance :

$V'$  = mean velocity of gas calculated for an equal flow through an empty tube of the same cross-section as the container, ft./sec. (cm./sec.).

$f$  = fractional voidage in the bed.

$D$  = average diameter of solid particles, ft. (cm.).

Furnas<sup>4.23</sup> gives a formula which is a complicated one, not in dimensionless group form. This formula needs an empirical constant for each material and it is not the same for heating as for cooling coke.

More recently Löf and Hawley<sup>4.24</sup> have measured the heat transfer to granitic gravel. They give a final result of

$$H_v = 0.79 \left( \frac{G'}{D} \right)^{0.7} \quad . \quad . \quad . \quad . \quad . \quad (4/14)$$

where  $H_v$  is the heat transfer coefficient in C.H.U./hr.-ft.<sup>3</sup> of bed volume- $^\circ\text{C.}$  ;

$G'$  is the air flow rate in lb./hr.-ft.<sup>2</sup> of bed cross-section ; and

$D$  is the equivalent spherical diameter of the particles in feet.

The formulæ of Löf and Hawley and of Saunders and Ford do not completely agree; Löf and Hawley discuss this discrepancy in

<sup>4.22</sup> SAUNDERS and FORD : "Heat Transfer in the Flow of Gas through a Bed of Solid Particles," *J.I.S.I.* (1940).

<sup>4.23</sup> FURNAS : *Bull. Bureau of Mines*, No. 307 (1929) ; No. 361 (1932).

<sup>4.24</sup> LÖF and HAWLEY : "Unsteady-state Heat Transfer between Air and Loose Solids," *Ind. Eng. Chem.*, **40**, p. 1061 (June, 1948).



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their paper. As a rule-of-thumb guide the expression should be used for roughly shaped material like broken granite, particularly in the size range around one inch, whilst the expression (4/13) should be used for more spherical bodies.

**4.2.4. Radiation.** (a) *General laws.* In the subject of radiation theoretical considerations play a much greater part than they do in conduction or convection. This is because radiation takes place in a perfect vacuum and hence its basic laws are independent of the properties of any particular substance. It thus becomes possible to apply the conceptions of thermodynamics to the calculation of the ideal radiation laws, making use of the fact that light exerts a pressure and hence can in theory be used in an ideal heat engine. The results of this thermodynamic study of radiation may be summarised as follows :

(i) A body at any temperature (apart from absolute zero) radiates heat as well as receiving it. When it is in equilibrium with its surroundings it radiates as much heat as it receives. In other cases the heat absorption is the difference between the heat received and the heat radiated. This is known as Prévost's theory of exchange.

(ii) Bodies radiate and absorb heat in proportion, i.e. the best radiator is also the best absorber.

(iii) A perfect body can be postulated as one which will absorb all the radiation falling on it. This is known as a "black body" because a matt black surface is the nearest approximation to it which can be obtained with a flat surface. A more perfect approximation can always be obtained by considering a small hole in a hollow body, the inside of the hollow being blackened and at a uniform temperature.

(iv) Bodies which are not black must reflect the radiant heat which they do not absorb unless they are sufficiently thin to transmit it. In all cases when the temperature of the body is not rising it follows from the first law of thermodynamics that the sum of the absorption, reflection and transmission is equal to the total heat falling on the surface. The emissivity of a body is defined as the ratio of the amount of heat radiated per unit area of the body to that radiated by a black body at the same temperature and the absorptive power is defined as the proportion of the heat falling on the surface of the body which is absorbed. From the second principle above, it follows that the emissivity of a non-black body is equal to its absorptive power. Hence, since by definition the



absorptive power of a black body is unity, its emissivity must also be unity.

(v) The total amount of heat radiated per unit time in all directions by unit area of a black body at an absolute temperature  $T^\circ \text{K}$ . is given by  $R = \sigma T^4$ . This law is known as "Stefan's Law" and the coefficient  $\sigma$  has the value

$1.01 \cdot 10^{-8} \text{ C.H.U./ft.}^2 \text{ hr. } ^\circ \text{K.}^4$  ( $1.73 \cdot 10^{-9} \text{ B.Th.U./ft.}^2 \text{ hr. } ^\circ \text{R.}^4$ \*  
or  $1.36 \cdot 10^{-12} \text{ cal./cm.}^2 \text{ sec. } ^\circ \text{K.}^4$ ) . . . . . (4/15)

when the temperature is in degrees absolute.

TABLE 4.2  
VALUES OF  $(T_k^4/100)$  FOR USE IN RADIATION CALCULATIONS

$T^\circ \text{C.}$	$(T_k^4/100)$	$T^\circ \text{C.}$	$(T_k^4/100)$	$T^\circ \text{C.}$	$(T_k^4/100)$
0	56	380	1820	860	16480
10	64	400	2050	880	17670
20	74	420	2305	900	18930
30	84	440	2585	920	20260
40	96	460	2885	940	21650
50	109	480	3215	960	23110
60	123	500	3570	980	24650
70	138	520	3955	1000	26260
80	155	540	4370	1050	30640
90	174	560	4815	1100	35540
100	194	580	5295	1150	41000
120	239	600	5810	1200	47080
140	291	620	6360	1250	53800
160	352	640	6950	1300	61220
180	421	660	7580	1350	69390
200	501	680	8250	1400	78340
220	591	700	8965	1450	88130
240	693	720	9725	1500	98820
260	807	740	10530	1550	110450
280	935	760	11390	1600	123070
300	1078	780	12290	1650	136750
320	1234	800	13260	1700	151530
340	1412	820	14270	1750	167490
360	1606	840	15350	1800	184670

Table 4.2 gives values of  $(T_k^4/100)$  which are useful for radiation calculations, since

$$R = 1.01 \left( \frac{T_k^4}{100} \right) \text{ C.H.U./ft.}^2 \text{ hr. or } 1.36 \cdot 10^{-4} \left( \frac{T_k^4}{100} \right) \text{ cal./cm.}^2 \text{ sec.}$$

Fig. (4)5 shows the radiation  $R$  as a function of  $T$ .

The total radiation from a non-black body having an emissivity  $E$  will be  $\sigma \cdot E \cdot T^4$ . If it is surrounded with other bodies at  $T_2$  and is

\* R here denotes degrees Rankine—the absolute Fahrenheit scale on which the melting point of water is  $492^\circ \text{R}$ .



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very small compared to its surroundings, the net heat transfer from these surroundings will be  $\sigma E(T_2^4 - T^4)$ . (The total radiation falling on it will be  $\sigma T_2^4$ , but the difference  $\sigma(1 - E)T_2^4$  will be reflected or transmitted). If, however, the radiated body is nearly as large as the surrounding objects and these also have an emissivity  $E_2$  which is appreciably less than 1, the net heat transfer will be

$$\frac{\sigma E E_2 \cdot (T^4 - T_2^4)}{E + E_2 - E E_2} \dots \dots \dots (4/16)$$

This formula can be used in the case of a partially transparent flame nearly filling a furnace whose walls and hearth are all at the same temperature. In practical cases where the walls and

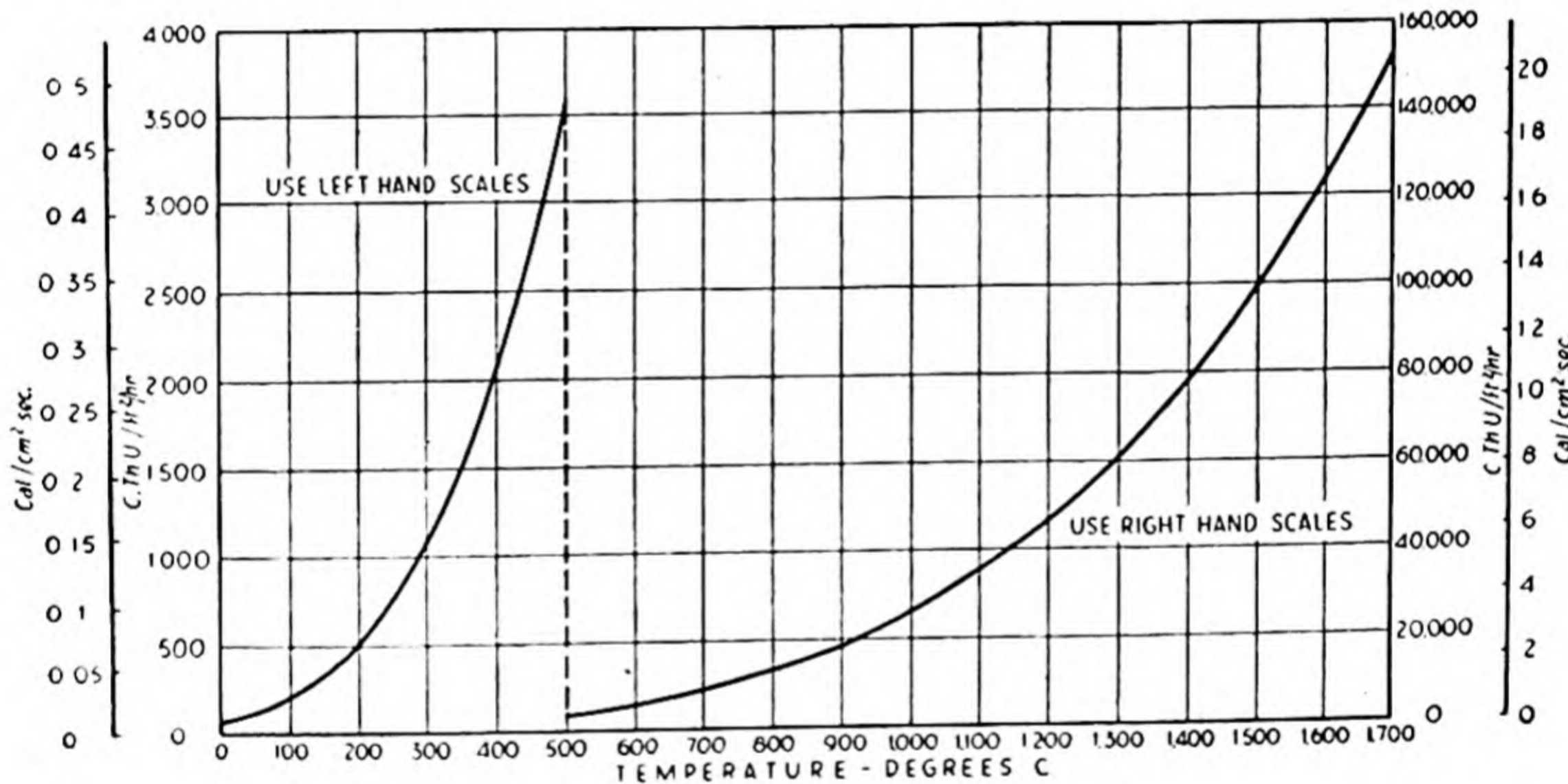


Fig. (4)5. Radiation from a Black Body.

hearth are at different temperatures it is usually necessary to make simplifying assumptions before the complete calculation is possible (see 4.3). The formula for the net heat transfer in the intermediate case where the body has area  $A$  and is surrounded by a larger hollow body of internal area  $A_2$  is given by

$$\frac{\sigma(T^4 - T_2^4)}{\frac{1}{E} + \frac{A}{A_2} \left( \frac{1}{E_2} - 1 \right)}$$

Table 4.3 gives some typical values of  $E$  for use in furnace calculations.

(vi) The radiation from a black body at any given temperature



is not confined to a single pure vibration, but has a widely distributed spectrum, i.e. it consists of energy spread over a number of wavelengths. The amount of energy at various wavelengths can

TABLE 4.3

EMISSIVITY OR ABSORPTIVITY OF VARIOUS MATERIALS AT VARIOUS TEMPERATURES (FROM DATA COMPILED BY M. FISHENDEN)

(Values expressed as percentage)

Substance	50° C.	500° C.	1000° C.
Building bricks	—	—	45
Stafford blue bricks	—	—	78
Fireclay brick	—	—	75
Corundum refractory brick	—	—	46
Sillimanite refractory brick	—	—	28
Silica refractory brick	—	—	66
Magnesite refractory brick	—	—	38
Natural concrete tiles	—	—	63
Painted surfaces, all colours	94	—	—
Aluminium paint	50	—	—
Lampblack, other blacks	96	96	96
Aluminium, polished	4	6	8
„ rough plate	7	—	—
„ oxidised heavily	20	33	—
Brass, polished	10	—	—
Brass, dull	22	—	—
Copper, polished	2	—	—
„ oxidised	60	88	—
Chromium, polished	10	28	38
Steel, rolled sheet	56	—	—
„ rough plate	95	98	—
„ heavily oxidised	88	98	—
„ calorised	79	79	—
Iron, cast	81	—	—
„ rough ingot	—	—	95
Nickel, plated on iron	5	—	—
Platinum wire	6	10	16
Tin, bright	6	—	—
Zinc, galvanised sheet	20	—	—
Alloy steel (8% Ni, 18% Cr)	—	35	—
Nichrome Wire, bright	65	71	79
„ „ oxidised	95	98	—
Glass	95	—	—
Water (0.1 mm. or more thick)	95	—	—

When the values given above are for a temperature beyond melting or combustion, they refer to absorptivity by the substance of radiation from a source at that temperature.

be measured by using a spectroscope in which quartz or other heat-transmitting prisms split up the radiation and the energy is measured by means of a heat measuring device such as a bolometer (resistance thermometer) single thermocouple or thermopile. The emission



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from a black body when plotted against the wavelength consists of a curve with a peak, falling off rather slowly above the peak and more sharply below it. The position of the peak moves towards the shorter wavelengths as the temperature rises, as well as rising very rapidly.

Non-black bodies may be "grey," in which case they will have a

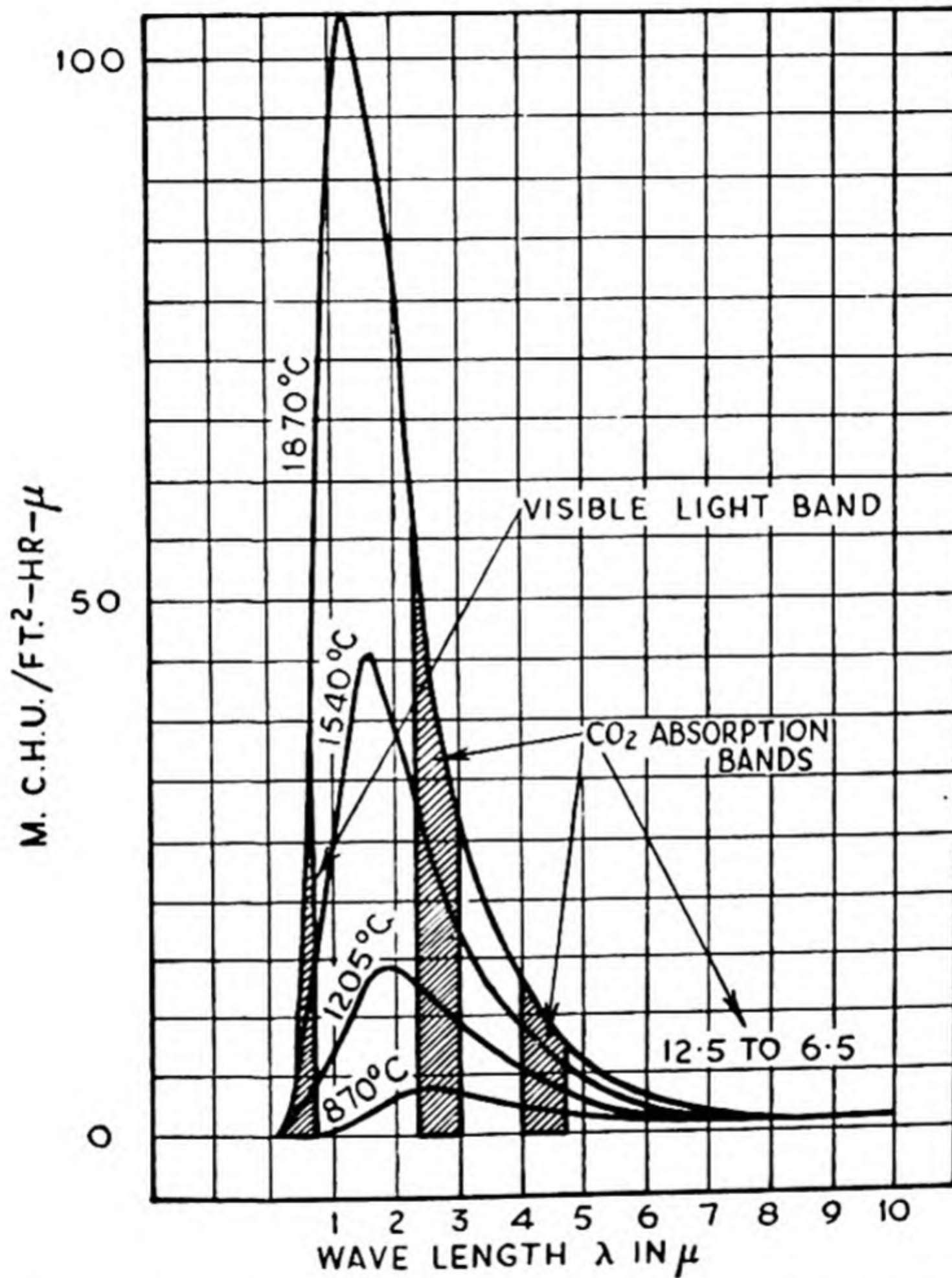


Fig. (4)6. Carbon dioxide absorption bands shown in relation to black body radiation as functions of wavelength. ( $\mu=1$  micron= $0.00003937$  inch.)

curve similar to that of the black body at the same temperature, but lying below it in an equal proportion at all wavelengths. Some bodies, however, are not "grey" but have a spectral distribution differing appreciably in shape from that of the black body *at the same temperature* although they cannot rise above it at any wavelength (unless a chemical reaction is proceeding). This property



can, however, be made use of where it is desired to obtain a maximum amount of radiation of a special wavelength.\*

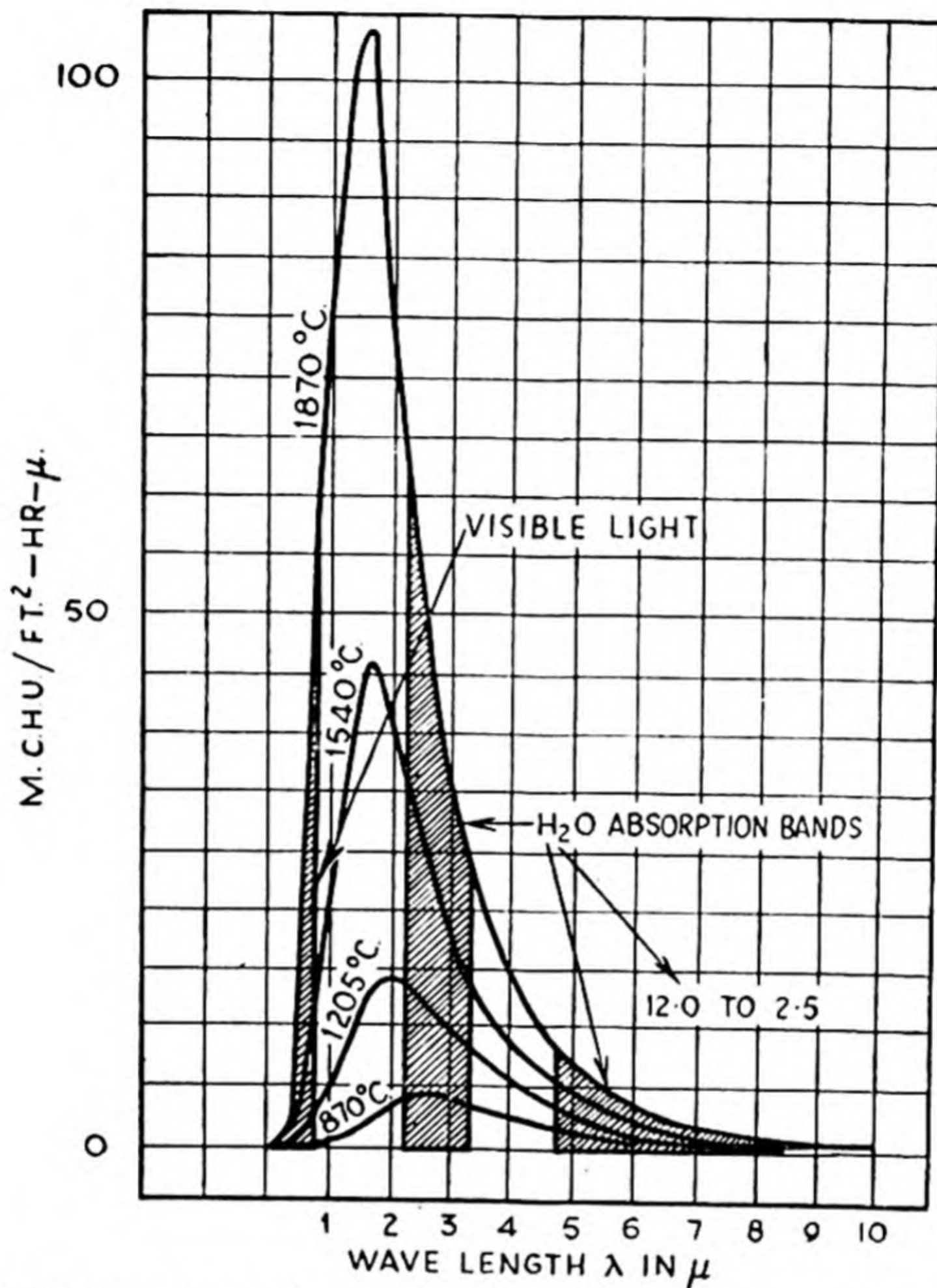


Fig. (4)7. Water vapour absorption bands shown in relation to black body radiation as functions of wavelength. ( $\mu=1$  micron= $0.00003937$  inch.)

(b) *Non-luminous gas radiation.* The triatomic gases  $\text{CO}_2$  and  $\text{H}_2\text{O}$  have radiation bands in the infra-red part of the spectrum. Consequently a layer of these gases which is sufficiently thick (see Figs. (4)6 and (4)7) will have a radiation curve which coincides

\* A body which gives nearly black-body radiation at the desired wavelength but considerably less at other wavelengths will reach a higher temperature than would a black body *supplied with the same amount of energy*, since in this case its total radiation must be the same. Hence the radiation from the non-black body at the favoured wavelength can lie above the value which would be obtained with a black-body in similar conditions. This is the principle of the use of thoria and ceria in the gas mantle.



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with the black-body curves over these parts of the spectrum, but remains zero in other parts of the spectrum. Since the amount of radiation which is absorbed in any given thin gas layer is proportional to the amount of radiation remaining at that point, it follows that one would expect a law for the emissivity or absorptive coefficient of the gas of the form  $\phi(1 - e^{-\alpha d})$ , where  $\phi$  is the emissivity of an infinitely thick belt,  $d$  is the thickness of the layer and  $\alpha$  is a factor depending on the effectiveness of the gas in absorbing wavelengths and on the pressure. Schack<sup>4.25</sup> showed in 1923 that such a formula does in fact apply. Since then the subject has been studied experimentally by (i) Schmidt,<sup>4.26</sup> who used jets of steam 6 cm. in diameter and increased the effective thickness by gilded mirrors; (ii) Hottel and Egbert<sup>4.27</sup> and Hottel and Mangelsdorf,<sup>4.28</sup> who passed preheated gases into an insulated tube with separate "guard-ring" chambers of preheated air at each end and measured the radiation of gas containing CO<sub>2</sub> and H<sub>2</sub>O in a wide range of compositions at temperatures up to 1050° C. and its absorption of radiation from a black body up to 1370° C.; (iii) McCaig,<sup>4.29</sup> who studied the effect of varying the total pressure, which he found to be real but small, so that in furnace systems where this total pressure is nearly atmospheric the effect can be neglected; (iv) Eckert,<sup>4.30</sup> who used apparatus similar to that of Hottel and Mangelsdorf but with less careful corrections. He worked up to a maximum gas temperature of 1300° C.

These experimental results have been reviewed by Lander<sup>4.21</sup> and by Schack.<sup>4.31</sup> Lander follows Hottel and Egbert in presenting the results in the form of families of curves, reproduced in Figs. (4)8 and 9, which can be used directly to estimate the radiation from combustion gases at a known temperature of a known thickness and

4.25 SCHACK, A.: *Z. Techn. Phys.*, **5**, p. 267 (1924).

4.26 SCHMIDT, E.: *Forsch. Ing.-Wes.*, **3**, p. 57 (1932).

4.27 HOTTEL and EGBERT: "Radiation of Furnace Gases," *Trans. A.S.M.E.*, **63**, p. 297.

4.28 HOTTEL and MANGELSDORF: "Heat Transmission by Radiation from Non-luminous Flames," *Trans. Amer. Inst. Chem. Eng.*, **31**, p. 517 (1935).

4.29 MCCAIG, M.: *Phil. Mag.*, Ser. 7, **34**, p. 321 (May 1943).

4.30 ECKERT, E.: "Measurement of the Total Radiation of Steam and CO<sub>2</sub> mixed with Non-radiating Gases at Temperatures up to 1300° C.," *Forsch.-Arb. Ing.-Wes. H.*, **387**, p. 20 (Berlin, 1937).

4.21 LANDER, C.: "Review of Recent Progress in Heat Transfer," *Proc. Inst. Mech. Eng.* (1942).

4.31 SCHACK, A.: "The Radiation of Furnace Gases," *Archiv.f.d. Eisenhüttenw.*, **13**, p. 241 (1939).



containing known percentages of  $\text{CO}_2$  and  $\text{H}_2\text{O}$ . To use these figures the value of the product  $pl$  ( $p$ =partial pressure of  $\text{H}_2\text{O}$  and  $\text{CO}_2$  in atmospheres which is the same as the fractional volume of

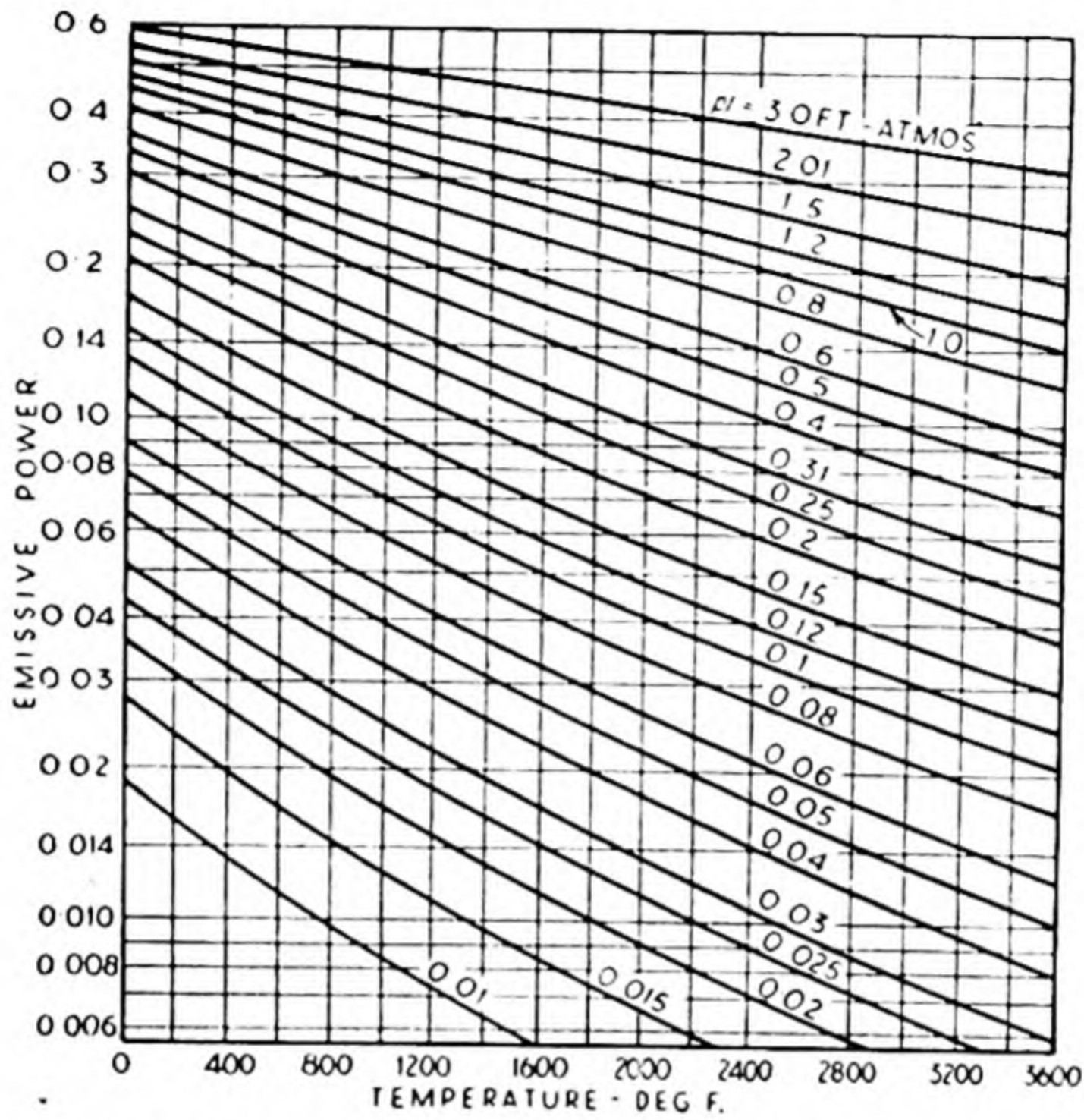


Fig. (4)8. Emissive Power of Carbon Dioxide. (After Hottel and Mangelsdorf).

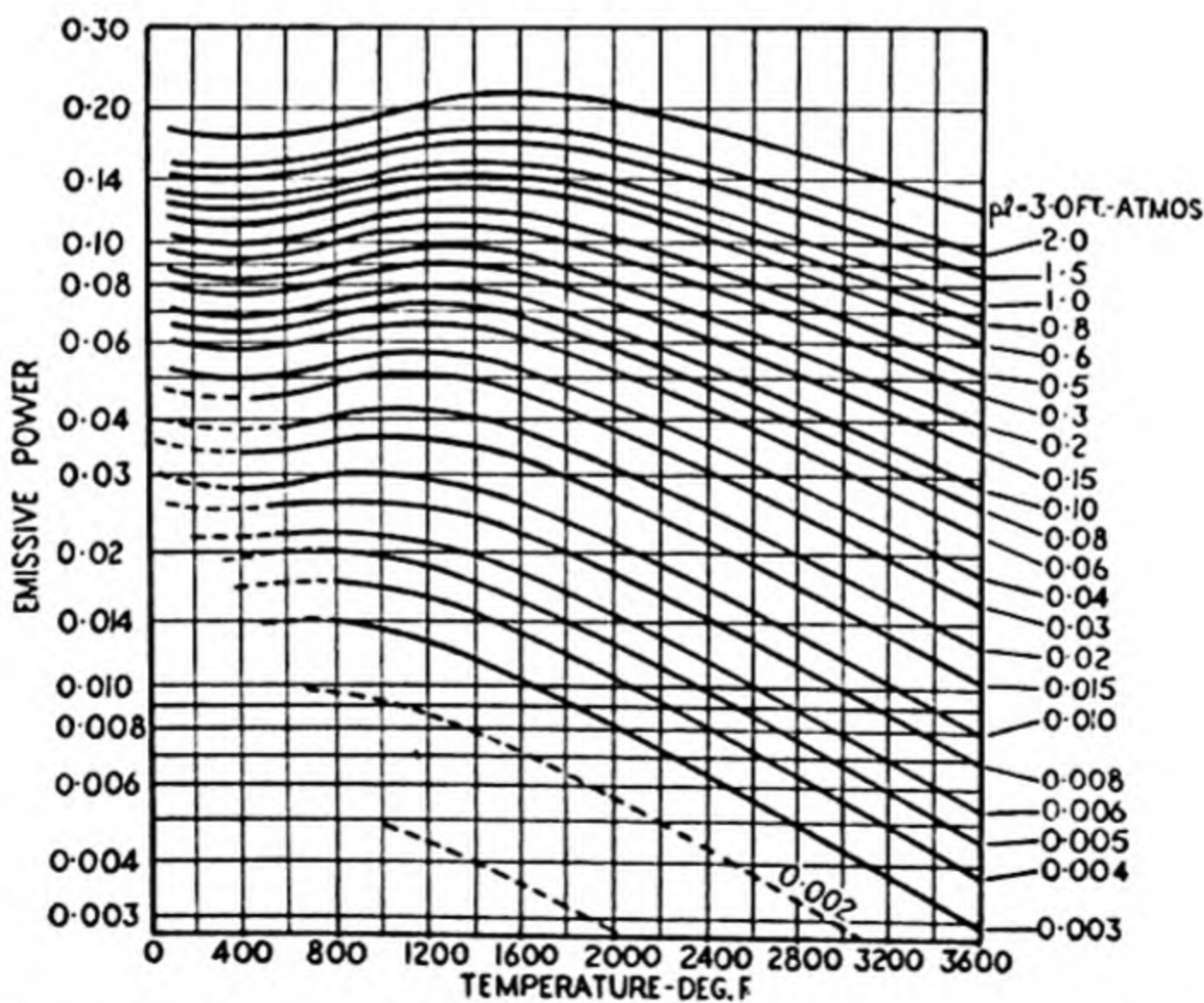


Fig. (4)9. Emissive Power of Water Vapour. (After Hottel and Mangelsdorf).



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these gases when the total pressure is atmospheric,  $l$  = thickness of flame in feet)\* is calculated and the emissivity  $E$  of the flame read from the curve corresponding to this value of  $pl$  at the known flame temperature. The total flame radiation due to the gas concerned is then  $\sigma ET^4$ . In the case of water vapour there is a pressure correction which is shown in Fig. (4)10. To use this curve the value of  $pl_{eff}$  is obtained from the figure corresponding to the actual

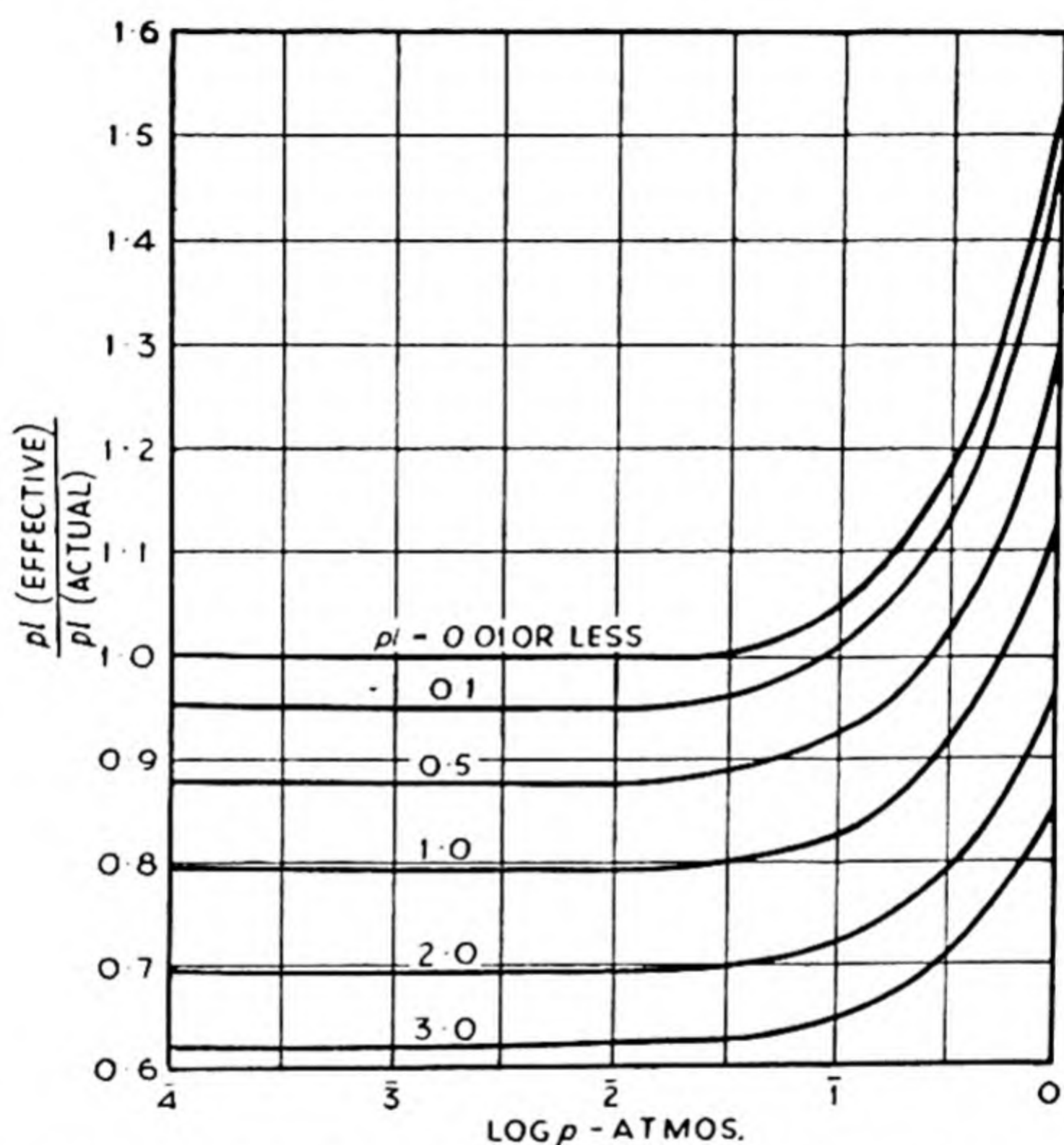


Fig. (4)10. Curves for finding "Effective" Value of  $pl$  from Actual Value of  $pl$  for Water Vapour Radiation. (After Hottel and Egbert.)

value of  $pl$  and of  $p$ , and this is used in Fig. (4)9. There is also a correction due to the interaction of the  $\text{CO}_2$  and the  $\text{H}_2\text{O}$ , which varies from zero at small values of  $pl$  to about 10% when  $pl$  is 3 ft.-atmospheres. Convenient tables, calculated from Hottel and Mangelsdorf, for three typical combustion gases are given below—Tables Nos. 4.4, 4.5 and 4.6. In furnace calculations where the wall emissivity is appreciably less than unity, formula 4/15 must be used, and the flame emissivity must be calculated either

\* The formulæ apply to a hemisphere of radius  $l$  of gas radiating to the centre, so that  $l$  is the same in all directions. Actual shapes are discussed below (see Table 4.7).



TABLE 4.4  
COMBUSTION GASES FROM PRODUCER GAS  
CO<sub>2</sub>—18% : H<sub>2</sub>O—8%  
Radiation in C.H.U./ft.<sup>2</sup> hr.

Temp. °C.	Thickness of gas layer in inches										
	1	2	3	4	6	8	12	16	24	32	40
200	22	31	39	47	59	67	84	98	116	130	145
400	86	123	153	184	224	261	314	370	439	489	543
600	234	357	430	505	604	709	848	988	1168	1300	1441
800	487	693	905	1067	1310	1526	1850	2120	2471	2768	3186
1000	836	1273	1554	1822	2250	2626	3162	3725	4476	5038	5710
1200	1219	1896	2362	2700	3360	3984	4900	5760	6864	7776	8832
1400	1598	2437	3156	3710	4554	5513	6790	7910	9510	10970	12780
1600	2090	3240	3956	5030	6275	7560	9324	10960	12920	15120	17390
1800	2497	3967	4980	5900	7390	8840	11280	13720	16540	18990	22560

TABLE 4.5  
COKE OVEN GAS  
CO<sub>2</sub>—9% : H<sub>2</sub>O=20%  
Radiation in C.H.U./ft.<sup>2</sup> hr.

Temp. °C.	Thickness of gas layer in inches										
	1	2	3	4	6	8	12	16	24	32	40
200	24	39	52	67	80	97	114	135	163	186	209
400	886	140	190	233	284	349	422	483	578	673	744
600	221	354	471	587	726	860	1080	1284	1557	1830	1958
800	448	729	992	1228	1500	1782	2170	2590	3120	3564	4334
1000	740	1190	1554	1956	2520	3000	3830	4583	5550	6540	7370
1200	1080	1795	2370	2880	3650	4460	5760	6910	8590	10180	11380
1400	1440	2333	3172	4155	5190	6270	7910	9910	12460	14700	17020
1600	1900	3020	4380	5540	6800	8320	10460	13230	17140	20790	22550
1800	2275	3760	5260	6960	8650	10640	13910	17480	21810	26410	30080

TABLE 4.6  
BLAST FURNACE GAS  
CO<sub>2</sub>—24% : H<sub>2</sub>O—4%  
Radiation in C.H.U./ft.<sup>2</sup> hr.

Temp. °C.	Thickness of gas layer in inches										
	1	2	3	4	6	8	12	16	24	32	40
200	20	30	36	41	49	56	67	78	92	109	118
400	89	124	148	165	194	228	270	311	355	410	458
600	254	350	422	479	564	645	755	848	964	1150	1290
800	548	771	919	1070	1237	1427	1674	1883	2106	2538	3160
1000	917	1319	1570	1836	2195	2506	2948	3350	3830	4500	5065
1200	1340	1987	2333	2760	3310	3734	4560	5155	5950	7060	7824
1400	1814	2693	3200	3795	4650	5440	6310	7170	8390	10230	11710
1600	2330	3475	4330	5190	6086	7230	8590	9800	11590	13990	15690
1800	2925	4380	5680	6790	7780	8990	10770	12410	14290	17860	19930



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directly from Figs. (4)8 and (4)9 or from these tables, and the value of  $(T_k^4/100)$  from Table 4.3.

Schack prefers to derive formulæ for the presentation of the experimental results and gives the following formulæ for the radiation from a hemisphere of gas of radius  $l$  to a point at the centre of the base :

$$h_{\text{CO}_2} = 0.48 \sqrt[3]{(p.l)(T/100)^{3.5}} \text{ C.H.U./ft.}^2 \text{ hr.}$$

(This formula can be used over the range  $600^\circ$ – $1350^\circ$  C., but is 28% too high at  $425^\circ$  C.).

Up to  $1000^\circ$  C. the values of Eckert are 20% lower than those of Hottel, while at  $1315^\circ$  C. the values of Eckert are 20% higher than those of Hottel).

$$h_{\text{H}_2\text{O}} = 3.5 p^{0.8} l^{0.6} (T/100)^3 \text{ C.H.U./ft.}^2 \text{ hr.}$$

(This formula can be used from  $400^\circ$  C., but the values of Hottel and Eckert disagree by amounts up to 30%, so it is not very reliable.)

In these formulæ

$h_{\text{CO}_2}$ ,  $h_{\text{H}_2\text{O}}$  are the coefficients for radiation from these gases ;  
 $p$  = partial pressure of  $\text{CO}_2$  or  $\text{H}_2\text{O}$  in fractions of an atmosphere ;

$l$  = radius of gas hemisphere in feet ;

$T$  = temperature of gas layer in  $^\circ$  K.

Where both  $\text{CO}_2$  and  $\text{H}_2\text{O}$  are present the combined radiation is 2–5% less than  $h_{\text{H}_2\text{O}} + h_{\text{CO}_2}$ .

When the surroundings are at a temperature somewhere near that of the gas it is convenient to use the simplified formula.

$$\alpha_{\text{CO}_2} = E_w \sqrt[3]{(p.l)(0.0070(t_g + t_w) - 4.13)} \text{ C.H.U./ft.}^2 \text{ hr. } ^\circ \text{ C.}$$

$$\alpha_{\text{H}_2\text{O}} = E_w p^{0.8} l^{0.6} (0.010(t_g + t_w) - 4.65) \text{ C.H.U./ft.}^2 \text{ hr. } ^\circ \text{ C.}$$

where

$\alpha_{\text{CO}_2}$ ,  $\alpha_{\text{H}_2\text{O}}$  denote the coefficients of heat transfer by radiation between the gas and the walls.

$E_w$  = emissivity of walls ; and

$t_g$ ,  $t_w$  = temperature of gas and walls in  $^\circ$  C.

When the gas is bounded on one side by the charge and on the other by a refractory crown, Schack considers that  $l$  in the formula should be made 25% greater than the actual thickness of the gas layer to allow for reflection by the crown. If the flame is completely sur-



rounded by heat-absorbing surface its effective thickness is increased by a factor of the order of  $1 + (1 - E_w)$ .

All these formulæ of Schack are clearly only applicable for limited values of the thickness  $l$ , since they give infinite radiation for infinite thickness, whereas clearly a flame of infinite thickness gives black-body emission for all the wavelengths in which it radiates and zero for all others, so that its emissivity remains a small fraction of unity. These formulæ of Schack indicate that  $\text{CO}_2$  radiation becomes significant for much thinner gas layers and is more strongly temperature sensitive than  $\text{H}_2\text{O}$  radiation.

It is necessary also to evaluate  $l$  in cases when the flame is not a hemisphere. Thus Schack considers that in an open-hearth furnace the effect of the crown is to increase the gas layer thickness by 25% ; if the roof were perfectly insulated the effect would be to double the gas layer thickness whatever the roof emissivity. Other cases are given in the following table.

TABLE 4.7

RADIUS OF HEMISPHERE EQUIVALENT TO VARIOUS GAS SHAPES

	$l$
Circular cylinder of height equal to diameter ( $d$ ) radiating to a point in the centre of a circular end	$0.77 d$
Circular cylinder very long, diameter $d$ radiating to a point on the cylindrical surface	$0.95 d$
Cube (side $a$ ) radiating to a point on the centre of one face	$0.67 a$
$\frac{1}{2}$ cylinder of infinite length radiating to a point on the axis	$1.26 r$
Sphere radiating to a point on the surface	$0.65 d$
2 infinite planes distance $h$ apart	$1.8 h$

Hottel and Egbert have shown that for industrial flames a reasonable approximation to  $l$  is 3.4 times the gas volume divided by the area of the boundary walls ; but the use of the table above is more reliable.

(c) *Luminous flame radiation.* Luminous radiation is a subject on which remarkably little scientific knowledge is available compared to its very great importance as a means of heat transfer in furnaces, particularly high-temperature ones. The radiation comes from very fine carbon particles (considerably less than  $1\mu$  in size) which are at a temperature (Schack<sup>4.32</sup>) very close to that of the surrounding gas and which are formed by a mechanism which is not yet fully understood ; it may be a polymerisation of carbon

<sup>4.32</sup> SCHACK, A. : *Z. Techn. Phys.*, 6, p. 530 (1925).



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atoms liberated in a partial combustion reaction or, as discussed by Rummel and Veh,<sup>4.33</sup> it may be a cracking of hydrogen from heavy hydrocarbons followed by burning of the hydrogen and adhesion of the residual hydrocarbons in pairs to form new larger molecules which again split off atomic hydrogen until carbon molecule "skeletons" are formed which become incandescent. Spectroscopic studies of soot-forming flames which are now being made may clear up this question, but it is also necessary to have considerably more knowledge of the radiating properties of these particles before formulæ can be given for calculating luminous radiation in practical cases. Rummel and Veh also showed that coke-oven gas could be rendered initially luminous by self-carburisation by preheating it to a temperature within a narrow range, a fact which may have considerable importance in the open-hearth furnace and glass tank furnace.

Since the carbon particles radiate at least to some extent at all wavelengths, luminous radiation from an infinitely thick layer will have an emissivity of 1, and a comparison with Figs. (4)8 and (4)9 show that where a flame is luminous the greater part of the radiation is likely to take place by this mechanism. Such further research on the radiating properties of the carbon particles in furnace flames and on the conditions of mixing of fuel with air and of the receiving of heat by the fuel which give various concentrations and sizes of carbon particles is therefore of the greatest importance.

If the carbon particles were large compared to the wavelengths of the radiation in which the bulk of the heat is transferred they could be treated as solid radiators and the emissivity of the flame would be simply the fraction of the total solid angle which is subtended by the particles at the given point. This emissivity is thus

$$E = 1 - e^{-nAl}$$

where  $n$  is the number of particles per unit volume ;

$A$  is the mean cross-sectional area of a particle ; and

$l$  has the same meaning as in the case of non-luminous radiation, namely, the radius of the radiating hemisphere.

Schack<sup>4.32</sup> showed, however, that the particles are so fine as to be almost transparent to radiation of the important wavelengths, so

<sup>4.33</sup> RUMMEL, K., and VEI, O.: "The Radiation of Luminous Flames," Pts. I and II, *Archiv. f.d. Eisenhüttenw.*, 14, pp. 489 and 533 (1941) (B.C.U.R.A. Translation Series No. 23).



that the formula for emissivity is considerably more complicated. In this case the particles will scatter light by the Rayleigh mechanism<sup>4.34</sup> and Senftlaben and Benedict<sup>4.35</sup> irradiated a luminous Bunsen flame with light from an arc lamp and made use of such scattered light to estimate the density of carbon in the flame and the number of particles. Mie<sup>4.36</sup> worked out the general formulæ for the case when the particles are conductors as well as dielectrics and when they have a size comparable to the wavelength concerned. Senftlaben and Benedict assumed the carbon particles were spherical, an assumption which has recently been checked by electron microscopy, and measured the scattered light with a spectrophotometer, the fraction scattered being about 4.4% of that falling on the flame. They deduced that the density of carbon in the flame was of the order of  $10^{-5}$  gm./c.c. and the number of particles about  $10^{-9}$ /c.c. The use of this method on industrial flames in conjunction with modern electron multiplier photocells is a promising tool. The same authors also developed<sup>4.37</sup> a method for temperature measurement in luminous flames based on the assumption that if the temperature of a wire in the flame is adjusted electrically until it just has no soot deposited on it, then it is at the flame temperature. As discussed already in 4.1.3, Hottel and Broughton found that the emissivity distance coefficient of luminous laboratory flames of amylacetate varied with the wavelength  $\lambda$  in the visible range in proportion to  $1/\lambda^{1.32}$  and based a two-colour flame temperature measuring method on this finding. They found it to vary as  $1/\lambda^{0.9}$  in the infra-red range. Such laws have not been checked for larger flames where temperature gradients are also liable to affect the applicability of such two-colour methods. Moreover, Becker<sup>4.38</sup> found for both the Hefner lamp and for soot an absorptivity  $\propto (1/\lambda^{1.4})$  which, although close to Hottel and Broughton's exponent, is sufficiently different to give appreciable errors. Becker deduced the density of carbon in his flame to be  $3.9 \times 10^{-6}$  by comparing the absorptivity of soot on a plate to that of the flame and also by placing a cold plate over the flame for a given time—he

<sup>4.34</sup> RAYLEIGH: *Phil. Mag.* (4), **41**, pp. 107 and 447 (1871); (5), **12**, p. 81 (1881).

<sup>4.35</sup> SENFTLABEN, H., and BENEDICT, E.: *Ann. der Physik.*, **60**, p. 297 (1919).

<sup>4.36</sup> MIE: *Ann der Physik.*, **25**, p. 327 (1908).

<sup>4.37</sup> SENFTLABEN, H., and BENEDICT, E.: "Eine Methode zur Temperaturbestimmung leuchtendenflammen," *Phys. Zeit.*, **19**, p. 180 (1918).

<sup>4.38</sup> BECKER, A.: "Über die Strahlung und Temperatur der Hefner lampe," *Ann. der Physik.*, **28**, p. 1017 (1909).



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concluded that 2.7% of the carbon was in the form of soot. Becker also measured the energy radiated by his flame as a function of wavelength, corrected it for each wavelength according to the emissivity deduced from measurements of the emissivity at this wavelength, and so obtained an ideal radiation curve for a black body at the same temperature as the flame. From this he deduced a flame temperature of 1670° K. (1400° C.) and then he also derived the temperature from measurements with an optical pyrometer on a single flame and on several in line behind one another so that he could deduce the effective absorptivity at the wavelengths used by the optical pyrometer; thus he obtained a second temperature of 1700° C., which is reasonable agreement. Further work on these lines would clearly be of value.

Some very interesting work on a laboratory luminous burner was carried out in Japan by Sakae Yagi.<sup>4.39</sup> He measured the absorption of radiation as a function of wavelength ( $\lambda$  microns) by soot layers on mica plates and by the flame using a small platinum-wound black-body furnace as the source and derived the value of the absorption coefficient  $k_\lambda(\text{cm.}^{-1})$  given by the formula

$$k_\lambda = \frac{C\omega}{\lambda} \left( 1 - \frac{0.15}{\lambda^2} \right)$$

where  $C$  is a constant depending on the burner conditions, which had the value 700–800 in his case; and  $\omega$  is the carbon content of the fuel gas in gm./cm.<sup>3</sup>

The emissivity of the usual hemisphere of flame for the wavelength  $\lambda$  is thus  $1 - e^{-k_\lambda l}$ . This formula is based partly upon a calculated application of Mie's theory as used by Senftleben and Benedict and partly on his experimental results that the radius of the soot particles is about  $0.1\mu$ , giving a maximum absorption at  $\lambda = 2\mu$  and that the number of soot particles is proportional to the number of grams of carbon contained in unit volume of the hydrocarbon fuel gas (acetylene or benzene).

In a later paper,<sup>4.40</sup> to which reference has already been made in 4.1.3 in connection with temperature measurement, Yagi showed that primary air reduces the temperature and number of the soot particles, while increase of hydrocarbon in the fuel gas lowers the

4.39 SAKAE YAGI: "Studies on Luminous Flames," *Jl. Soc. Chem. Ind. Japan*, **40**, No. 2, p. 50b (1937).

4.40 SAKAE YAGI and SUROKIE KAWAI: "Studies on luminous Flames," Pt. III, *Jl. Soc. Chem. Ind. Japan.*, **40**, No. 4, p. 144 (1937).



flame temperature because of the increased emissivity and hence increased radiation to the cold surroundings.

The radiation from luminous flames of industrial size has been studied experimentally in special flame tunnels by a number of workers, including Lindmark,<sup>4.41</sup> whose primary concern was with boilers; Sherman<sup>4.42</sup> and Trinks.<sup>4.43</sup> Lindmark used a vertical water-cooled cylinder subdivided longitudinally into six sections, the heat transfer to each of which could be measured separately, and fired down the axis with oil and pulverised-coal flames. He concluded that with oil flames the bulk of the radiation was from separately formed soot particles and not from the original oil droplets, a conclusion in line with the discussion of the combustion of an oil droplet given in Chapter 3, since the oil droplets cannot get hot enough to radiate before they evaporate. Pulverised coal, on the other hand, is much more dependent on the fuel particles for radiation, since these can be coked and heated to very high temperatures before they burn. In general oil flames were found under these cold wall combustion conditions to radiate a slightly greater percentage of the calorific value of the fuel to the walls before leaving the chamber than did pulverised coal flames, and oil flames had a much more intense radiation in the very early part of the flame.

Sherman built a horizontal firebrick cylinder with internal length 14 ft. and 3 ft. 6 in. internal diameter, with small holes every 2 ft. for measurement of the flame temperature (bare platinum thermocouples) and gas composition and larger holes permitting the insertion of a thermopile and a block water-cooled target to measure the radiation from various thicknesses of flame. The rate of heat input was about 1,500,000 C.H.U./hr. The emissivity of the whole flame thickness was deduced from the gas temperature measurements with the platinum thermocouples, but it was not found to be possible to calculate the emissivity of flames of any other thickness from the readings with variable thickness because these latter did not fit the equation

$$E = E_{\infty}(1 - e^{-kl})$$

<sup>4.41</sup> LINDMARK, T., and EDENHOLM, H.: "The Flame Radiation in Water-cooled Boiler Furnaces," Pt. I; LINDMARK, T., and KIGNELL, E.: "The Flame Radiation in Water-cooled Boiler Furnaces," Pt. II; LINDMARK, T.: "Study on Heat Transmission in Boiler Furnaces," *Ingeniors Vetenskops Akademien*, Nos. 66, 109 and 91 respectively.

<sup>4.42</sup> SHERMAN, R. A.: *Trans. A.S.M.E.*, **56**, pp. 177, 401 (1934).

<sup>4.43</sup> TRINKS, W., and KELLER, J. D.: "Tests of Radiation from Luminous Flames," *Trans. A.S.M.E.*, **58**, p. 203 (1936).



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a result also found by Mayorcas<sup>4.44</sup> on open-hearth flames of producer gas. The emissivity of the whole flame was constant along the furnace at about 20% for rapidly burnt natural gas and for a "semi-luminous" natural gas flame while a fully luminous natural gas flame gave a value around 60% and a pulverised coal flame fell steadily from 60% at the burner to 40% at the outgoing end. In the second paper Sherman describes comparative tests on flames of pulverised coals of various particle sizes in the same furnace, and concludes that while the temperature and hence the total radiation are affected by fineness of grinding, excess air and rate of firing, the emissivity is only affected to a marked degree by changes in the type of coal, the highest values being obtained with the slowest burning of the three coals which had 40–44% volatile.

Trinks and Keller<sup>4.45</sup> also used a brick-lined combustion tunnel for their experiments on luminous radiation, and in their case it was about 10 ft. long and 2 ft. square. They measured the flame temperatures with sheathed thermocouples and the radiation with a total radiation pyrometer sighted across the whole flame width at a water-cooled box behind a slit in the opposite wall. The natural gas flame was produced in a tunnel at the end similar to a glass tank port. It was shown that an emissivity of 95% could be obtained at the beginning, falling to 75% 10 ft. along, using a special type of burner called a cracking and mixing burner in which part of the gas is injected with a high velocity just before the flame leaves the port mouth for the main tunnel. With the conventional port the best emissivity was only just over 50%, falling to 38% at the outgoing end.

To summarise the work on luminous radiation it is clear that the concentration and size of carbon particles formed in a hydrocarbon flame and giving it luminosity are functions of the rates of arrival of heat and oxygen into a small elementary volume of gaseous hydrocarbon as it passes through the furnace, but that the fundamental work which will elucidate the character of these functions, and so enable the effect of luminous radiation in industrial furnaces to be calculated, has not yet been adequately carried out.

Wohlenberg<sup>4.45</sup> has carried out extensive calculations on the luminosity of pulverised coal flames assuming that this luminosity

4.44 MAYORCAS, R.: "The Emissivity of Producer-gas Flames in Open-Hearth Furnaces," *I.S.I. Special Report*, No. 37, Pt. 12, p. 129 (1946).

4.45 WOHLBERG, W. J., and WISE, G. E.: "Distribution of Energy in the Pulverised Coal Furnace," *Trans. A.S.M.E.*, p. 531 (1938).



arises from the presence of the solid particles of coal and ash. As fairly good agreement is obtained with experimental values, this would appear to indicate that the luminosity due to the combustion of the volatiles does not play a large role in these cases.

### 4.3. Heat Transfer in the Furnace Chamber

**4.3.1. Separation of the Heat Transfer Calculation from the Complete Complex Process.** In a furnace system the temperature of the gases depends on many factors, one of which is the heat transfer itself. On the other hand, all the formulæ for heat transfer discussed under the preceding section imply a knowledge of the temperature of the heating gases ; consequently, the complete calculation of the process involves the solution of two sets of simultaneous equations, one set of which give the flame temperature at any point as a function of the heat content of the gases (the combustion energy balance), which in turn depends on the preheat and combustion rate, while the other gives the heat transferred as a function of the flame temperature at each point.

To see the problem clearly it is necessary to distinguish between :

- (i) *The design and operation variables*, which include the surface areas and shapes of the walls and charge and the quantities of fuel and air inserted ;
- (ii) *the intermediate variables*, which are mainly of interest as steps in the calculation of the process (these include the gas temperature at any given point, the number of carbon particles in a luminous flame and the completeness of combustion at any given point) ; and
- (iii) *the performance variables*, of which the really important ones are (a) the heat transfer rate to the material on which output depends, and (b) the heat loss from the system and the heat carried out by the combustion gases as percentages of the fuel energy, on which the thermal efficiency depends.

Three methods of tackling this complex problem are available, each of which has something to recommend it. (1) The simplest method is to assume certain values of the intermediate variables (e.g. a mean gas temperature of  $T^{\circ}\text{C.}$ , an effective emissivity of 50% and combustion proceeding uniformly throughout the furnace chamber) and then to calculate the heat transfer rates and other performance variables from these assumptions. This method of calculation really implies that the derivation of the intermediate



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variables from the given design variables needs no calculation because it can be obtained from experience of similar furnace systems. It is accordingly open to the objection that it is inflexible in regard to the calculations of new types of furnace system. Examples of such assumptions are where Trinks<sup>4.46</sup> proposes that the average temperature difference to be used in calculating the heat-transfer rates to a block of steel inserted cold in a hot furnace and brought up nearly to the furnace temperature should be taken as one-third of the initial temperature difference between the furnace and the stock. Similarly, in open-hearth furnace calculations it is customary to assume an effective temperature of about 1800° C. because it is likely that the silica crown would not survive if the flame temperature were higher than this. This method is subject to the very considerable defect that it is only applicable to furnaces closely similar to those on which the assumptions have been found to work, i.e. it does not make full use of the theoretical knowledge available. For example, it is clearly desirable to obtain a flame of high emissivity, but the performance of a furnace with such a flame would not appear to show any advantage if calculated from assumptions obtained from furnaces with flames of low emissivity. Calculations of this type have been carried out for the open-hearth furnace in the second part of a recent paper<sup>4.47</sup>. In this paper it is assumed that a flame of emissivity  $e$  fills the space between the open-hearth roof and the slag surface, both of which are effectively black. It is proved that the net heat flow to a roof of normal thickness (i.e. the heat conducted through the roof) is very small compared with the gross radiation reaching it, so that only 1/100 of the radiation which reaches it is conducted through it, while 99/100 are reradiated or reflected. When the net radiation falling on the slag covering the steel, which has a surface temperature of either 1400° C. or 1600° C., and the resulting roof temperatures are calculated for various values of the flame temperature and emissivity, the result is as shown in Fig. (4)11. From this figure it will be seen that for a given furnace and a given slag surface temperature the downward heat flow, and hence the net heat to the steel, is very closely correlated with the roof temperature whatever the flame emissivity, so that if the roof temperature is fixed, the downward heat flow is also very nearly fixed and for each flame emissivity

<sup>4.46</sup> TRINKS: *Industrial Furnaces*, 1, p. 39.

<sup>4.47</sup> THRING, M. W.: "General Theory of Heat Transfer in the Open-Hearth Furnace," *I.S.I. Special Report*, No. 37, p. 171.



there is a corresponding permitted flame temperature. On the other hand, such difference in downward heat flow as does occur for a given roof temperature is in the direction that a lower emissivity and a higher flame temperature give a lower roof temperature for a given emissivity than does a higher emissivity associated with a lower flame temperature. It can be concluded that when the heat is insufficient to raise the roof temperature to the maximum consistent with safety it is very desirable to increase the emissivity of the flame provided this can be done without reducing the flame temperature, but that when the maximum roof temperature has

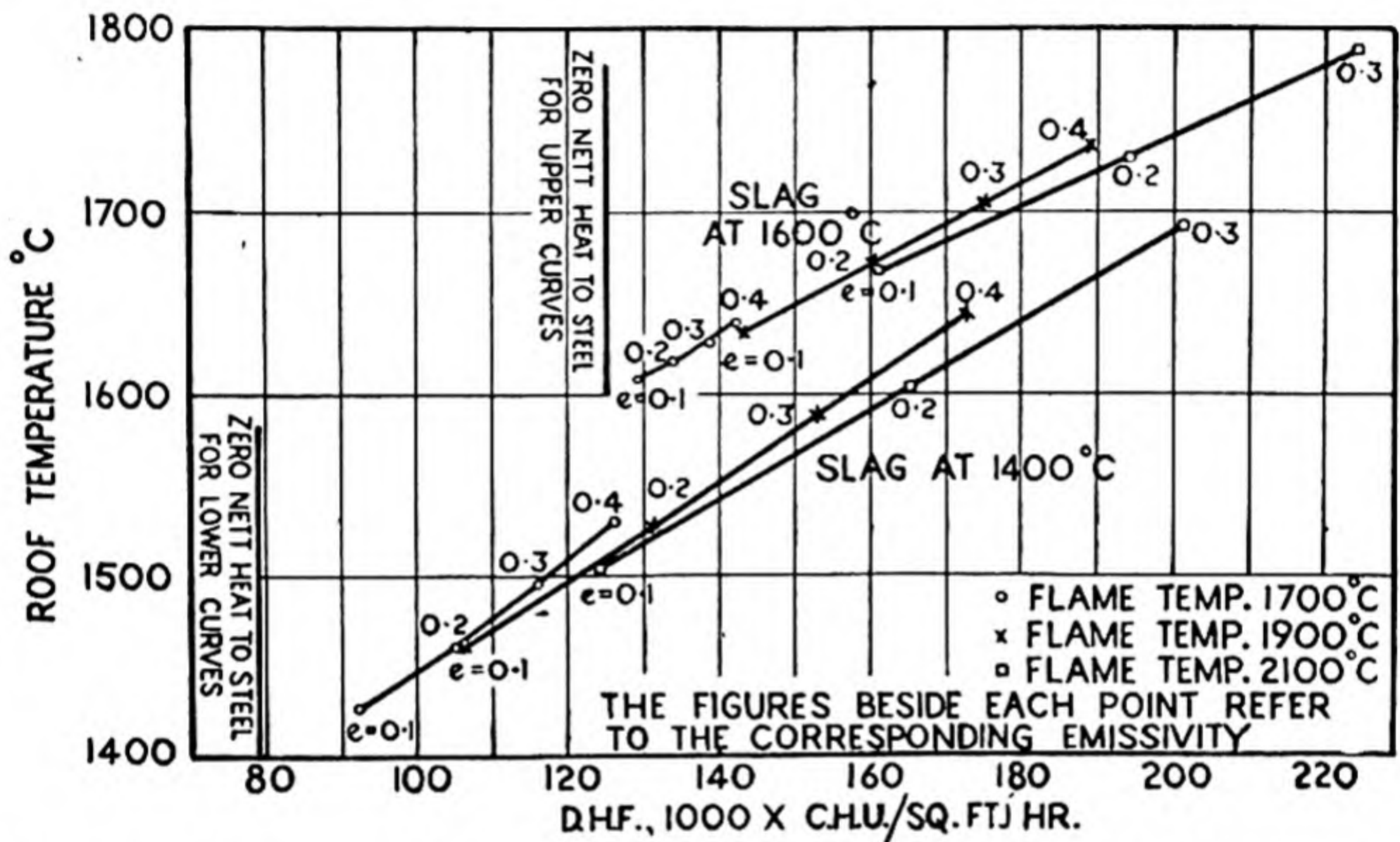


Fig. (4)11. Calculated relations between Open-Hearth Furnace Roof Temperature and Downward Heat Flow for Constant Flame Temperature and for Constant Flame Emissivity.

been reached it may in some cases be better to reduce the emissivity of the flame rather than its temperature.

(2) It is possible, as mentioned above, to write down the simultaneous equations governing (a) intermediate variables as a function of the design and performance variables (e.g. the equation giving the gas temperature in terms of preheat and amount of heat transfer to the charge), and (b) the performance variables as a function of the design and intermediate variables (e.g. the heat transfer to the charge as a function to the gas temperature and flame luminosity). These equations can, in theory, then be solved both for the intermediate and the performance variables. In general this method of calculation is too complicated for practical use, but it is of great



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value in regard to the estimation of the principal alterations which will result from a new design. McAdams<sup>4.48</sup> gives typical calculations of this type in the two limiting cases of instantaneous combustion and combustion proceeding uniformly throughout the furnace chamber, i.e. he partially simplifies the problem by assuming that we have complete knowledge of the combustion conditions while the heat transfer and temperature conditions remain to be calculated. Wohlenberg<sup>4.47</sup> has carried out similar calculations for the pulverised coal flame. For this method of calculation the gas

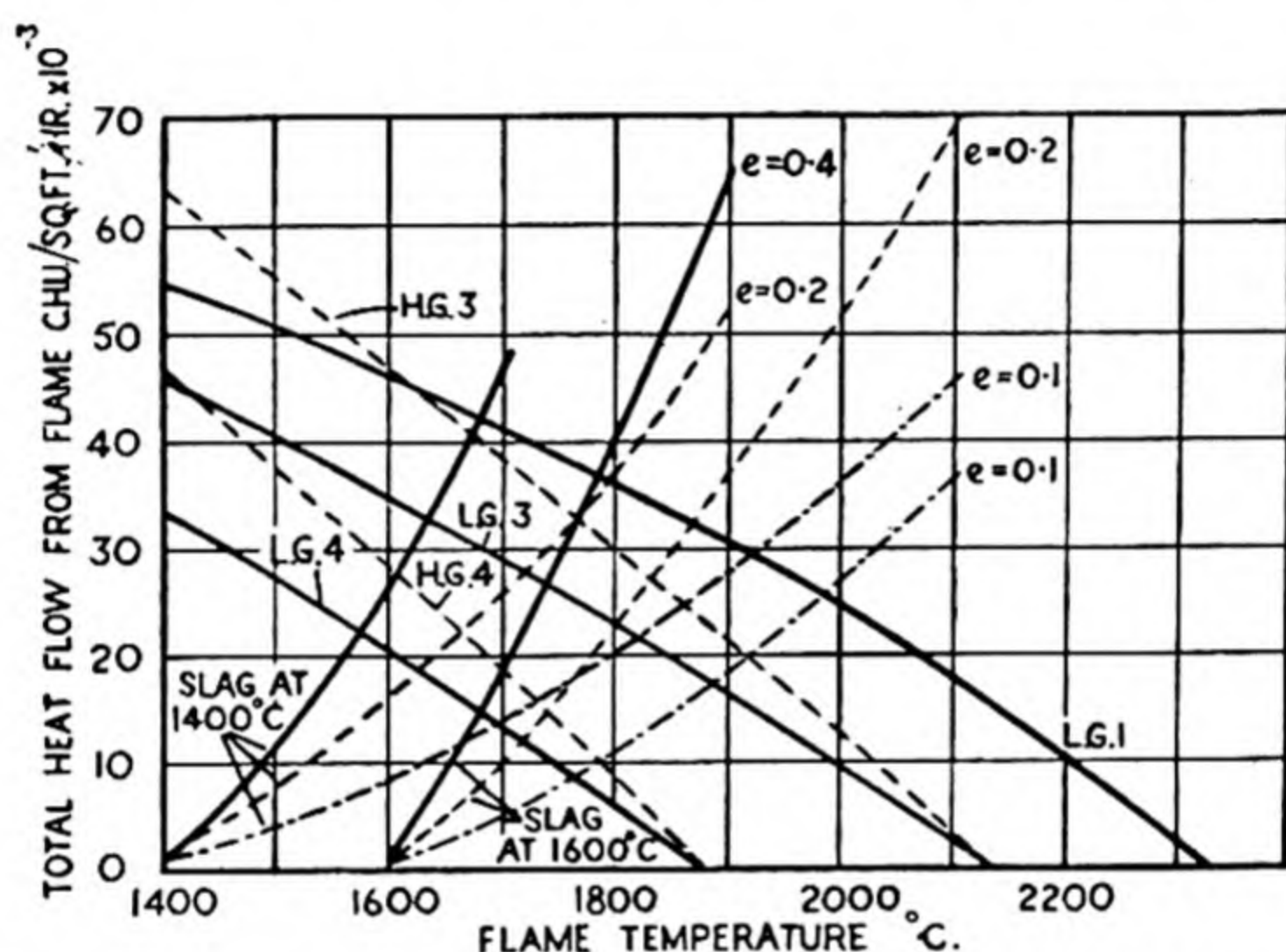


Fig. (4)12. Graphical method of evaluating heat transfer to the charge in an open-hearth furnace for a given Emissivity and Gas/Air Ratio, but unknown Gas Temperature.

temperature is obtained by calculating the heat content of combustion gases in C.H.U./ft.<sup>3</sup> and then using the Rosin-Fehling IT diagram as discussed in Section 2.1.2.

This method of solving simultaneous equations for the radiation from the flame and the heat available in it has been used in the third part of the paper referred to above<sup>4.47</sup> to evaluate the heat which will be transferred to the steel under various combustion conditions. A graphical method of carrying out this evaluation is shown in Fig. (4)12. This shows that the factor which most greatly affects the heat flow to the steel is not the increase in heat input nor increase in flame emissivity, but rather increase in the theoretical flame temperature by reducing the excess air or raising the CO<sub>2</sub> in the waste gases.

4.48 McADAMS, W. H.: *Heat Transmission*, pp. 81-88 (McGraw Hill, 1942)



Heiligenstadt<sup>4.49</sup> has developed a method which is of great practical value for the calculation of the probable performance of a counterflow pusher-type billet-heating furnace with gas firing. This method is equivalent to the solution of the simultaneous equations, but is actually carried out by means of successive approximation. Suppose the throughput and dimensions of a furnace are decided and it is desired to calculate the fuel consumption, temperature-distance curve for the charge, and gas-leaving temperature. The first step is to calculate the gas-leaving temperature using the following purely empirical formula :

$$\text{Flue temperature} = 625 + 3.3b \cdot (1 + 1/a) - 300a \text{ } ^\circ \text{C.}$$

where  $b$  = throughput in lb. of steel per ft.<sup>2</sup> of hearth area per hr.  
 $a$  = fraction of the hearth covered by steel.

Similarly, the combustion velocity is calculated from the probable combustion coefficient of the burner, using the formula given in 3.3 and a guess is made of a curve for the charge temperature with position along the furnace using the fixed values for the temperature of the charge entering and leaving. So far, therefore, the method has been simply an application of empirical formula obtained with similar furnaces, and if no more were done it would be open to the criticism given to the methods of the first type discussed above. Actually, however, Heiligenstadt only used these figures as the starting-point for more detailed calculation. He divides up the hearth longitudinally into sections, the first being that from the burner to the point where the steel leaves the furnace, while the remainder divide the path of the steel into equal areas. Detailed calculation of the amount of combustion taking place in the first section where there is no steel and of the temperature of the gas at the end of this section is then made, using the formula of 3.3 and allowing for wall heat losses, which are calculated from the mean furnace temperature. A similar calculation is made for the first section of the hearth except that here the heat transfer to the charge is also allowed for, knowing the temperature of the gas entering the section and using the assumed temperature of the charge in this section. In this way the gas temperature at the end of the second section is calculated. The calculations are repeated for the whole length of the furnace, thus obtaining a curve for the gas temperature as function of distance along the furnace. Using these gas

<sup>4.49</sup> HEILIGENSTADT, W.: *Warmstechnische Rechnungen für Bau u. Betrieb von Ofen*, Chapter VIII (Dusseldorf, 1925).



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temperatures a revised set of curves for the heating up of the blocks is then obtained and if necessary the whole process is repeated.

A similar method of calculation is given by Heiligenstadt for estimating quantitatively the effect of such variables as particle size on the heating in a shaft furnace. Here again the fuel consumption is assumed to be known and an estimate of the gas exit temperature is made from the fuel consumption. The heat content of the charge descending the column, including that absorbed by chemical reactions, is plotted as a function of temperature, and the temperature of the gas corresponding to the transfer of each unit of this heat is then plotted on the same curve, allowance being made for delayed combustion of the gas. The total heat transfer is then divided up into sections in each of which the gas temperature, batch temperature and batch heat capacity are regarded as constant. The heat transfer coefficient  $\alpha$  in each region is calculated by Heiligenstadt from Furnas's formula (Section 4.2.3) and the time of heating in each section is obtained from the formula

$$t = \frac{g \cdot c}{\alpha \cdot r} \log_e \frac{T_g - T_c}{T_g - T_o} \text{ hr.} \quad (4/17)$$

where  $g$  = weight of charge per unit surface area lb./ft.<sup>2</sup> ;

$r$  = factor for conductivity of charge ;

$c$  = specific heat of charge  $\left( \frac{\text{C.H.U.}}{^\circ\text{C}-\text{lb.}} \right)$  ;

$T_g$  = gas temperature in section,  $^\circ\text{C.}$  ;

$T_o$  = temperature of charge entering section,  $^\circ\text{C.}$  ;

$T_c$  = temperature of charge leaving section,  $^\circ\text{C.}$

(3) It is also possible to express experimental results on the heat transfer from flames as a function of the design variables only, omitting the intermediate variables altogether. This would be done ideally by the methods of similarity with dimensionless criteria as discussed in 4.2.2. The most important dimensionless criteria concerned are :

- (i) the ratio of the length along the flame to a length which can be built up from the heat transfer data ;
- (ii) the ratio of this latter length to the combustion length ;
- (iii) the emissivity of the flame ; and
- (iv) the ratio of the radiation of the actual gases to those from a flame of equal emissivity at the theoretical flame temperature for complete adiabatic combustion with the given fuel-air



ratio. This ratio  $= \frac{\sigma T_f^4}{\sigma T_i^4}$ , where  $T_f$  is the actual flame temperature and  $T_i$  the theoretical flame temperature for complete combustion without heat loss, both expressed on the absolute centigrade scale ( $^{\circ}$  K.).

It would thus appear that it would be ideally possible to give dimensionless formulæ for general flame radiation which could be calculated under any conditions thermally similar to the conditions of the experiments. Such a set of dimensionless curves has been produced in connection with the radiant heat absorption in boiler firing,<sup>4.50</sup> but the data have so far proved inadequate for constructing them for furnace firing owing, partly, to the essential difference from boilers that in furnaces the ratio of heat absorption temperatures to theoretical flame temperatures is of the order of 1 : 2, whereas in boilers it is 1 : 6 to 1 : 10. This third method of calculation, like the first, can only be applied to systems similar to those on which the experiments were done. When furnace design reaches the point, however, where it can be used, the prediction of results will become an accurate science and not a combination of experience and art. It has, however, proved possible to apply similarity theory to the application of model results to geometrically similar full-scale furnaces in certain cases.

It can be shown<sup>4.51</sup> that the dimensionless number, whose equality in model and original ensures that the fraction of the heat input which is radiated to the charge is the same, is

$$N' = \frac{H}{\sigma E(T_a^4 - T_b^4)L^2} \quad \dots \quad (4/18a)$$

where  $H$  = time-rate of heat input C.H.U./hr. to whole furnace ;

$\sigma$  = Stefan's constant ;

$E$  = effective emissivity of charge, wall and flame ;

$T_a, T_b$  = absolute temperature of flame and surroundings ;

$L$  = characteristic length.

If  $T_a/T_b$  is the same in model and original,  $N'$  reduces to

$$N' = \frac{H}{\sigma E(\theta^4)L^2} \quad \dots \quad (4/18b)$$

<sup>4.50</sup> HURVICH, A. M. : *Bull. de l'Acad. des Sciences de l'U.S.S.R., Classe des Sciences Techn.*, p. 23 (1943). See also RUDORFF : *Engr. Boiler House Rev.*, p. 86 (April 1943).

<sup>4.51</sup> THRING, M. W. : "Application of Similarity Principles to Thermal Transport Systems," *Nature*, **159**, p. 203 (8 Feb. 1947).



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where  $\theta$  is the absolute temperature at any position in model and original.

If combustion is governed by the rate of turbulent mixing the criterion that combustion shall have proceeded to the same relative extent at the same fraction of the distance along model and original is the equality of the Reynolds number

$$R_e = \frac{VL}{\nu}$$

where  $V$  is a characteristic velocity at a chosen point in model and original ;

$L$  is a characteristic length for either ;

$\nu$  is the kinematic viscosity of the fluid at the point which is most important for deciding the mixing rate, e.g. in the issuing producer gas jet in an open-hearth furnace.

If both  $N$  and  $R_e$  can be made the same and heat transfer by radiation and combustion by mixing are the major processes taking place, then good similarity can be obtained. When these criteria involve incompatible conditions it is necessary to find a way of obtaining approximate similarity, and it is shown in 4.52 that this can be done in a model open-hearth furnace either (1) by changing the cross-sectional area of the gas port by a different factor from that used for the areas of the furnace chamber so that the  $L$  factor in  $R_e$  is different from that in  $N$ , or (2) changing the effective wall temperature so that  $T_a^4 - T_b^4$  is different in model and original, or (3) adjusting the calorific value of the gaseous fuel in the model so that the requisite velocity is obtained to give the same  $R_e$  while at the same time the calorific heat input is adjusted to give the right value of  $N$ . Leckie<sup>4.53</sup> made use of the second of these three methods. For example, if the scale of the model is a quarter of that of the original, then Reynolds number similarity is satisfied by multiplying the velocities by 4 so that the quantity of fuel going in ( $H_o$ ) is reduced to one quarter. Without change of temperatures this would increase  $N$  to four times its previous value—in other words, radiation would not cool the flame sufficiently.  $N$  is then reduced by operating with relatively cold walls, i.e. by reducing the ratio of  $T_b$  to  $T_a$  in equation 4/18a so that the bracket in the

4.52 THRING, M. W.: "The Construction of Models in which more than One Process is similar to the Original," *Proc. Inst. Chem. Eng.* (1948).

4.53 LECKIE, A. H., HALL, J. R., CARTLIDGE, C., ALLEN, J. F., and FENTON, G.: "An Experimental Furnace for the Investigation of Open-Hearth Conditions," *J.I.S.I.*, 155, Pt. 3 (March 1947).



denominator is increased. In practice a water-cooled hearth is used instead of slag at  $1600^{\circ}\text{C}$ . and then the thermal resistance of this hearth is adjusted by placing on it thin refractory plates of a thickness determined by experiment. In this way independent control of  $R_e$  and  $N$  is achieved.

**4.3.2. The Radiation from Actual Furnace Flames.** The calculation of non-luminous radiation when the gas temperature is known, by the methods outlined in 4.2.4, is sufficiently accurate to require no further work. On the other hand, as discussed in 4.2.4, even the mechanism of luminosity is not fully understood and there is a long way to go before the actual number and size of the carbon particles produced in a flame under any given set of design variables can be calculated. As far as actual furnaces are concerned, however, all that can be done is to give rough empirical figures whereby

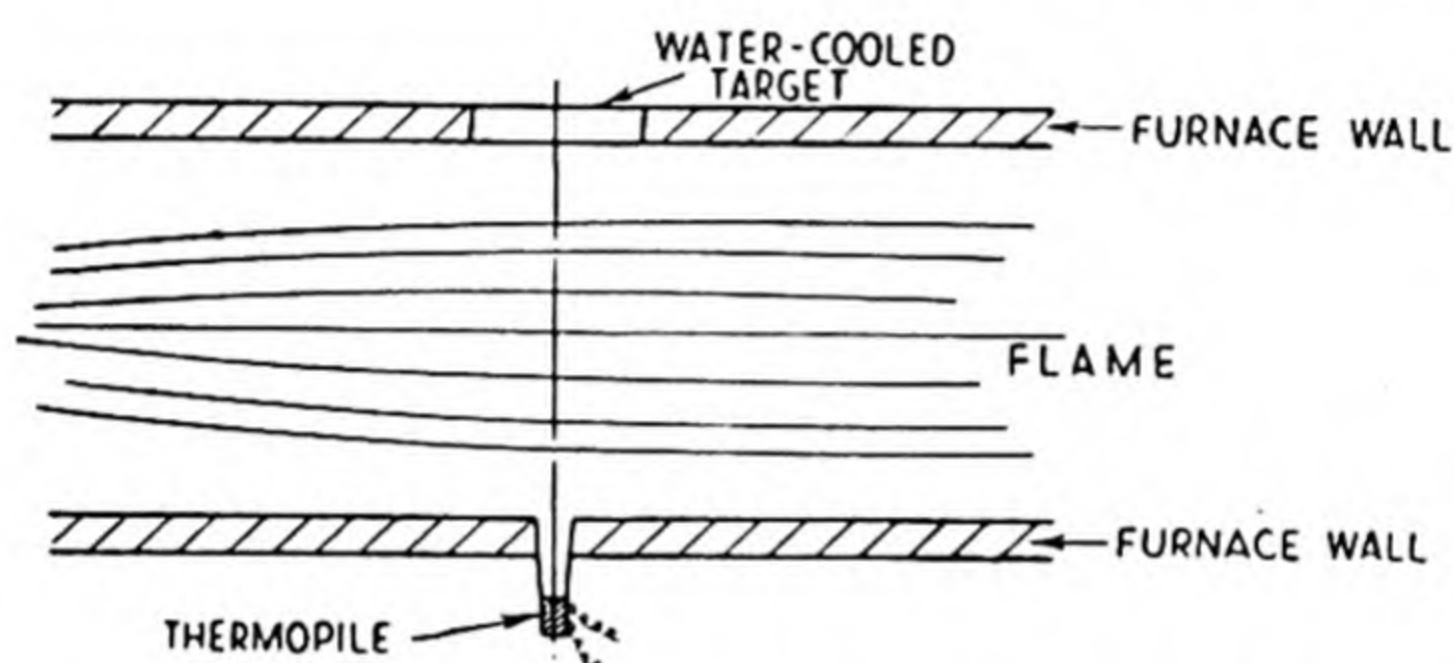


Fig. (4)13. Wall-type Radiometer for Furnaces.

the flame radiation can be guessed at if the temperature is known. In order to give these figures we must consider the methods by which they are obtained.

Two ways of *measuring* radiation from flames of size comparable with those in furnaces have been developed. These may be called the Wall method and the Heat-flow Meter method respectively.

*The Wall method* consists of the insertion of a small heat-absorbing element in a hole in the wall of the furnace arranged with water-cooled screens so that it can "see" a small amount of the furnace gases, Fig. (4)13. Radiation from the furnace wall on the other side of the gases is eliminated by using a non-reflecting water-cooled screen. This method gives the true radiation from the flame, although it does not make it possible to separate the effects on this radiation of flame temperature and emissivity. Such a separation can only be made if the flame temperature is also measured by methods given in 4.1.3 or if two more readings are



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taken, namely, the radiation of the flame with a hot surface behind it and the radiation of this hot surface alone. This method was used by Lent<sup>4.54</sup> to verify Schack's formula for non-luminous gases, and has also been used to a considerable extent in boiler tests.<sup>4.55</sup> It was also used in the tests on luminous flames by Sherman<sup>4.42</sup> and Trinks,<sup>4.56</sup> the results of which, as far as our knowledge of luminous radiation is concerned, have been given in 4.2.4. Other methods of measuring flame radiation in furnaces, especially that of Schmidt,

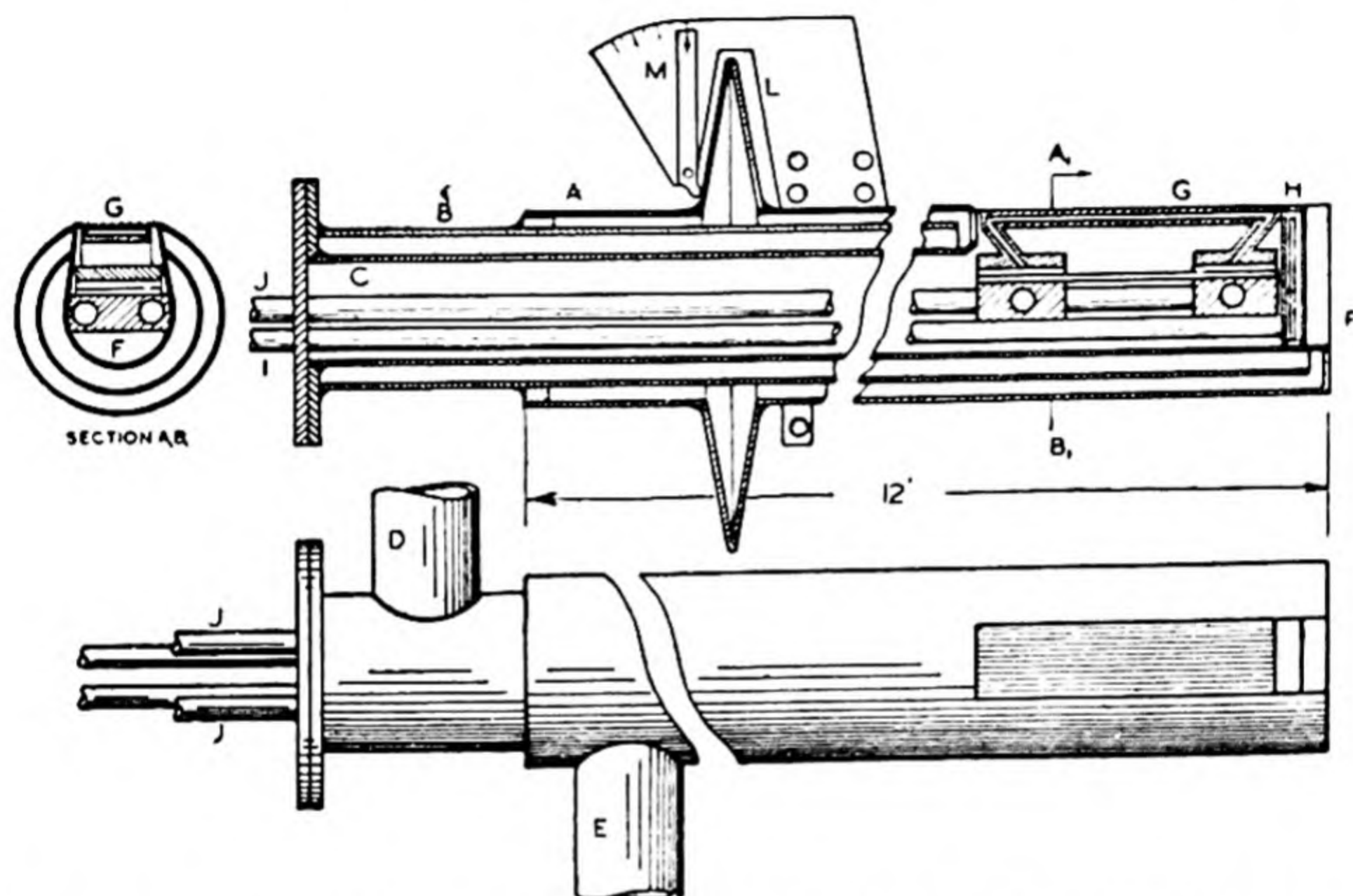


Fig. (4)14. Heat-flow Meter for Use in Furnaces. (*Baulk and Thring.*)

are closely bound up with the measurement of flame temperature and have been discussed in 4.1.3.

The *Heat-flow Meter* type of instrument consists of a calorimeter on the end of an arm which can be placed at any appropriate point in the furnace and the heat transfer to it measured, Fig. (4)14. In its ideal form this instrument consists of a calorimeter surrounded by a guard ring and so arranged that the heat transfer to a small plane section from the whole hemisphere surrounding it can be measured. By facing the calorimeter in diametrically opposed

4.54 LENT: Mitt. 65 of the *Wärmestelle*, Düsseldorf.

4.55 WOHLBERG, W. J., and MULLIKEN, H. F.: "Review of Methods of Computing Heat Absorption in Boiler Furnaces," *Trans. A.S.M.E.*, p. 531 (1935).

4.56 TRINKS, W.: "Tests of Radiation from Luminous Flames," *Trans. A.S.M.E.*, p. 203 (1936).



directions the net heat transfer through the plane can be calculated. Details of various types of heat-flow meter are discussed in *Coal Research*.<sup>4.57</sup> This type of instrument is not so suitable as the wall type for measuring the radiation of the flame alone, as it is not easy to separate the radiation from the flame from that of the walls behind it. It is, however, an instrument of very great value for probing the conditions in actual furnaces, as it can be inserted to any desired point to study the uniformity along or across the flame and to compare one furnace or fuel with another to study flame widths and lengths, etc.

The relative merits of non-luminous and luminous flames in actual furnaces have been much discussed, the argument in favour of the former being that they allow the radiation from the crown to reach the charge better than in the case of luminous flames. From the thermodynamic point of view developed in 4.1.1 clearly the higher heat-transfer coefficient associated with luminosity is very desirable provided the refractories are not overheated thereby. Sherman<sup>4.58</sup> concludes that the luminosity of the flame has an effect in increasing the heat transfer rate in open-hearth furnaces where the crown and the stock have the same temperature, and in boilers where all the walls are heat absorbing, but has little effect where there is a crown much hotter than the charge. In making the comparison it is, however, very important to be clear as to what is to be varied in going from one type of flame to the other. Thus if the aim is to bring the gas temperature as near to that of the charge as possible with a fixed rate of heat transfer in the furnace, and luminosity can be obtained without loss of combustion efficiency, there is no doubt that its higher emissivity must be of value. Similarly when it is necessary to obtain a good heat transfer to the charge with a limited roof temperature as in the open-hearth furnace, an opaque flame with a temperature gradient (i.e. hotter near the charge) is of value. On the other hand where the temperatures of the luminous flame are considerably lower than those of the non-luminous flame owing to the slowing down of combustion necessary to cause the flame to be luminous, or where the aim is to obtain as high an output as possible regardless of fuel efficiency, a non-luminous flame may be advantageous.

4.57 BAULK, R. H., and THRING, M. W.: "The Heat-flow Meter and the Distribution of Heat in Furnaces," *Coal Research*, p. 19 (June 1944) also *J.I.S.I.*, No. 1 (1946).

4.58 SHERMAN: *Trans. A.S.M.E.*, p. 183 (1934).



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In designing a furnace for heating solid materials the angle subtended by the flame as seen from the charge is clearly of great importance, as is the fraction of the surroundings of the flame which are covered with the cold charge. Alterations in these factors may produce as great an increase in the heat absorption rate as will increased flame luminosity, without the added disadvantage of increased refractory troubles which is liable to occur with more strongly radiating flames. This may be seen in terms of the argument of Section 4.1.1, where it will be seen that increase of  $A$  is as effective as increase of  $K$ .

**4.3.3. Convection in the Furnace Chamber.** Many furnace designers use very much simplified formulæ, such as that of Etherington,<sup>4.59</sup> according to which the heat transfer coefficient due to convection  $\alpha_c$  is 1 C.H.U./ft.<sup>2</sup> hr. °C. Similarly, Trinks gives a figure of 2 in these units, Heiligenstadt gives the value  $\alpha_c=4-10$  and the formula  $\alpha_c + \alpha_r = 10 + 0.06(T_g - 700)$  for the sum of convection and radiation in pusher furnaces and the figures of Table 4.8 for this sum in batch furnaces.

TABLE 4.8  
HEAT TRANSFER COEFFICIENTS IN BATCH FURNACES  
C.H.U./ft.<sup>2</sup> hr. °C.  
Temperature of Furnace, °C.

Temp. of Charge °C.	1300	1200	1100	1000	900
0	76	64	52	46	40
200	73	61	50	44	38
400	70	58	48	42	36
600	68	55	46	39	33
800	65	52	44	37	31
1000	63	49	42	34	—
1200	60	—	—	—	—

It is quite clear that these formulæ cannot have widely extended value since, as shown in Section 4.2.3, convection is proportional to the gas velocity to a power between  $\frac{1}{2}$  and 1, whereas these formulæ assume it is independent of the velocity. Formulæ such as those of Section 4.2.3 are much more reliable, since they are based on the use of dimensionless magnitudes, the main difficulty being to decide which of the simplified arrangements for which such formulae have been obtained corresponds most closely to the actual conditions.

Where heat is to be largely by convection, as at low temperatures

<sup>4.59</sup> ETHERINGTON: *Modern Furnace Technology*, p. 299.



and with non-luminous flames, it is very desirable to have the flame passing between the parts of the charge or underneath it, as well as along the top.

No data are available for the effect of flame impingement on convection, although some approximate idea can be obtained by comparing the coefficients for heat transfer at right angles to the outside of a tube with that corresponding with the inside of the tube with the same velocity. Convection to a particulate charge permeated by the gases, as in the producer-gas-fired shaft kiln, is best calculated from the formulæ of Furnas or, better, by the more recent and simpler treatment given by Löf and Hawley.<sup>4.24</sup> Some experiments have also been carried out on convection to falling bodies, such as those which occur in a rotary kiln, but these results have not yet been reduced to the dimensionless form in which they could be applied with confidence.

**4.3.4. The Effect of Muffles on Heat Transfer in the Furnace Chamber.** When a muffle is introduced between the flame and the charge, the heat transfer rate for a given temperature difference is very greatly reduced or, alternatively, it is necessary to raise the flame temperature very much to obtain the same heat transfer. The muffle interferes both with convection and, what is perhaps not so generally realised, with radiation. As far as convection is concerned, the effect can be calculated by finding a thermal resistance between the flame and the outside of the muffle, the thermal resistance of the muffle itself and that between the inside of the muffle and the charge. Since the heat transfer through each of these resistances is the same it is possible to calculate both it and the temperature concerned by the same method, as will be discussed in the next subsection (4.3.5) for the heat loss from furnace walls. If the gas inside the muffle is subject to natural convection only, the thermal resistance inside will be very great. At higher temperatures, e.g. over 1000° C. the muffle acts rather as an interruptor of radiation, and in this case it reduces the net heat transfer  $H$  for a given temperature difference between the flame and the charge to one half or less. Thus in the best possible case where the walls have emissivity unity and the muffle is a perfect conductor, the heat transfer equations are

$$H = \sigma(T_1^4 - T_2^4) = \sigma(T_2^4 - T_3^4)$$

<sup>4.24</sup> LÖF and HAWLEY: "Unsteady State Heat Transfer Between Air and Loose Solids," *Ind. & Eng. Chem.*, **40**, p. 1061 (June 1948).



#### 4.3.4 THE SCIENCE OF FLAMES AND FURNACES

so that, eliminating the muffle temperature  $T_2$ ,

$$H = \frac{\sigma}{2}(T_1^4 - T_3^4) \text{ C.H.U./ft.}^2 \text{ hr.} \quad (4/19)$$

where  $T_1$  is the mean radiant temperature of flame backed by outside walls ; and

$T_3$  is the charge temperature (both in  $^{\circ}\text{K.}$ ).

These calculations show that the price in fuel which must be paid for the chemical benefit of keeping the combustion gases from the charge can be very high, since a poor heat transfer coefficient means a high gas leaving temperature and hence a high fuel consumption.

**4.3.5. The Calculation of Heat Transfer in Parallel and Counter-flow Heating.** In both parallel and counterflow heating the temperatures of the heating gas and of the heated substance vary continuously from one end to the other, so that the temperature difference available for heat transfer depends upon the heat transfer in the neighbouring sections. The step-wise method of Heiligenstadt for calculating counterflow pusher-type and shaft-type furnaces has already been discussed in 4.3.1 ; in cases where combustion is complete before the heat transfer starts, a fuller mathematical solution of the simultaneous equations is possible. Such a solution is of particular value in the calculation of recuperators, but may be applied to other cases also. The notation used here corresponds, however, to the case where a gas is being heated through a wall ; where a solid is being heated it is only necessary to replace  $V'c_p'$  by heat capacity of the solid passing a given point in 1 hour (C.H.U./ $^{\circ}\text{C. hr.}$ ). Fig. (4)15 illustrates the state of affairs in the two types of heating system. It is assumed that the rate of heat transfer at a chosen point is given by  $\alpha(T - T')$  C.H.U./ft.<sup>2</sup> hr. where  $\alpha$  is a combined heat transfer coefficient (not necessarily independent of temperature),  $T$  is the temperature of the heating gas, and  $T'$  that of the heated substance at this point. It is also assumed that a certain fraction,  $1 - \eta$ , of the heat given up by the heating gas at any point is lost through the walls,  $\eta$  being transferred to the charge.  $\eta$  is assumed constant along the system.

With these assumptions the basic equations for the heat transfer in a small element of the counterflow system are

$$V \cdot c_p \delta T = -\alpha(T - T')\delta A = \frac{V' \cdot c_p' \cdot \delta T'}{\eta}$$

if  $\delta A$  is the element of heating surface measured from the hot end.

\*  $V'$ ,  $V$  are the N.T.P. volume flows of heated and heating gas, and  $c_p'$   $c_p$  their specific heats per unit N.T.P. volume.



These equations can, if  $\alpha$ ,  $\eta$ ,  $V'c'_p/Vc_p$  are independent of temperature, be readily integrated and the boundary conditions for the end temperatures inserted, to give

$$\text{Final heated gas temperature } T_1' = \frac{1}{a - z_0} \{T_1(1 - z_0) + T_2'(a - 1)\}$$

$$\text{Final heating gas temperature } T_2 = T_1 - a(T_1' - T_2') \quad . \quad . \quad (4/20a)$$

where  $a = \frac{V' \cdot c'_p}{\eta V \cdot c_p}$  and

$$z_0 = \exp\left(\frac{-\eta \alpha A_0(a - 1)}{V'c'_p}\right)$$

$A_0$  being the total area.

In most calculations it is quite sufficiently accurate to take  $\alpha\eta$  and

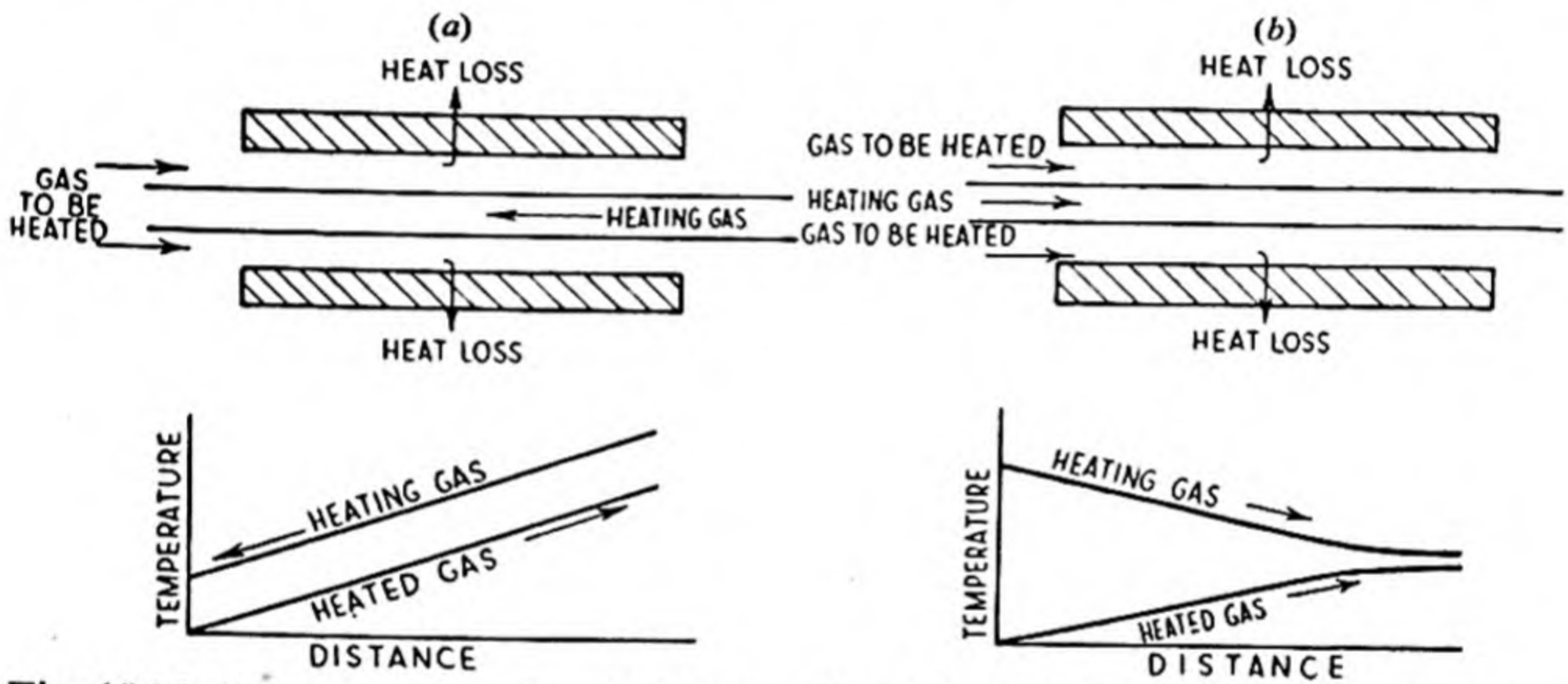


Fig. (4)15. Diagrams showing (a) Counterflow Heat Exchanger and (b) Parallel-Flow Heat Exchanger.

$V'c'_p/Vc_p$  as constant at the mean values, but if this is found to involve too large an error the following procedure can be adopted. The temperature distribution along the preheater is calculated using the formulæ

$$T' = \frac{1}{a - 1} (T_1(z - 1) + T_1'(a - z))$$

$$T = T_1 - a(T_1' - T') \quad . \quad . \quad . \quad . \quad . \quad (4/20)(b)$$

where  $z$  is the same function as  $z_0$  but with the total area  $A_0$  replaced by the partial area  $A$  measured from the hot end, assuming constant coefficients at the mean values. The whole heating surface is then divided up into sections and these sections recalculated separately using the coefficients corresponding to the calculated mean value in each section.







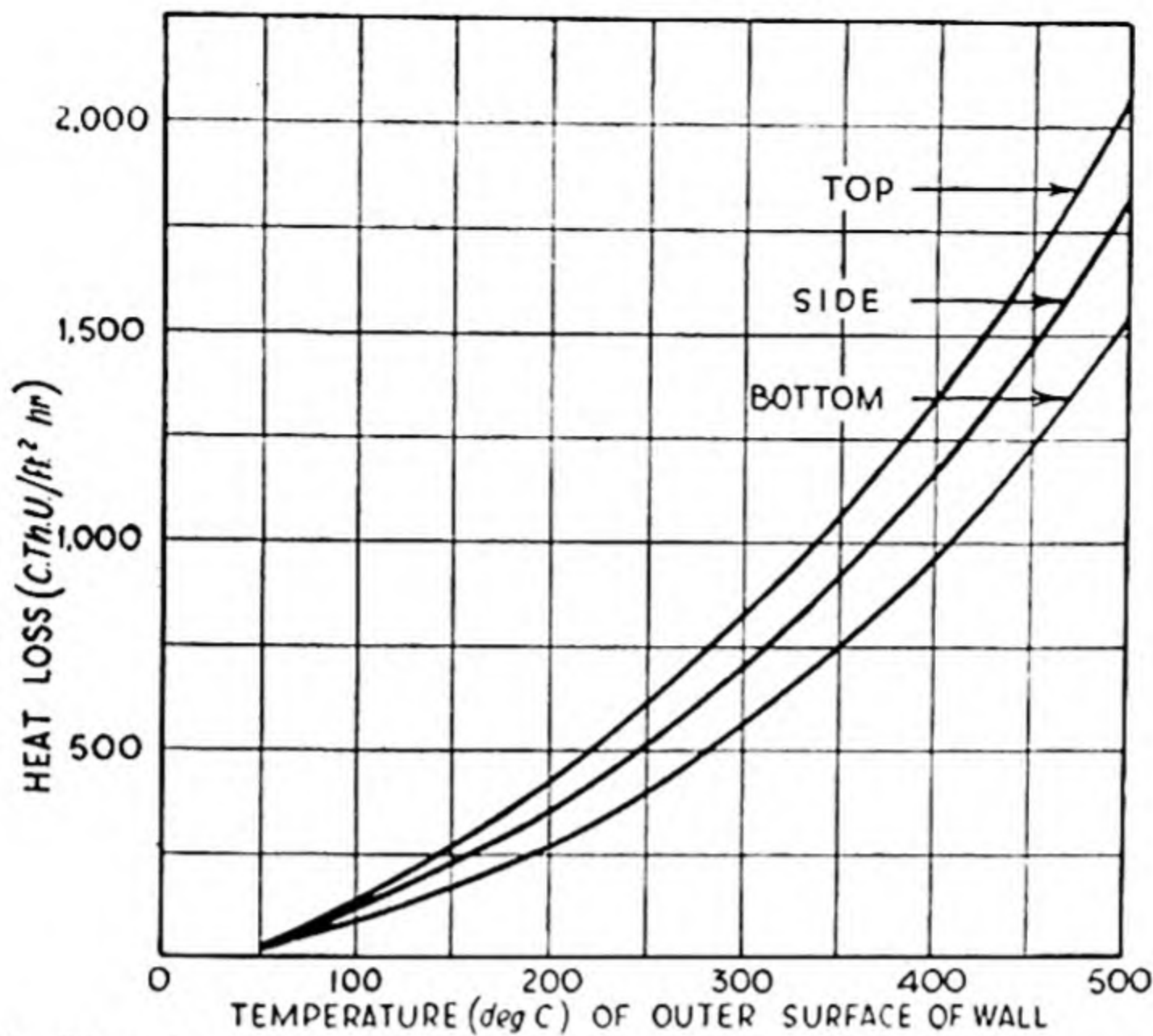


Fig. (4)16. Heat Loss from Furnace Walls Plated and Aluminium Painted.

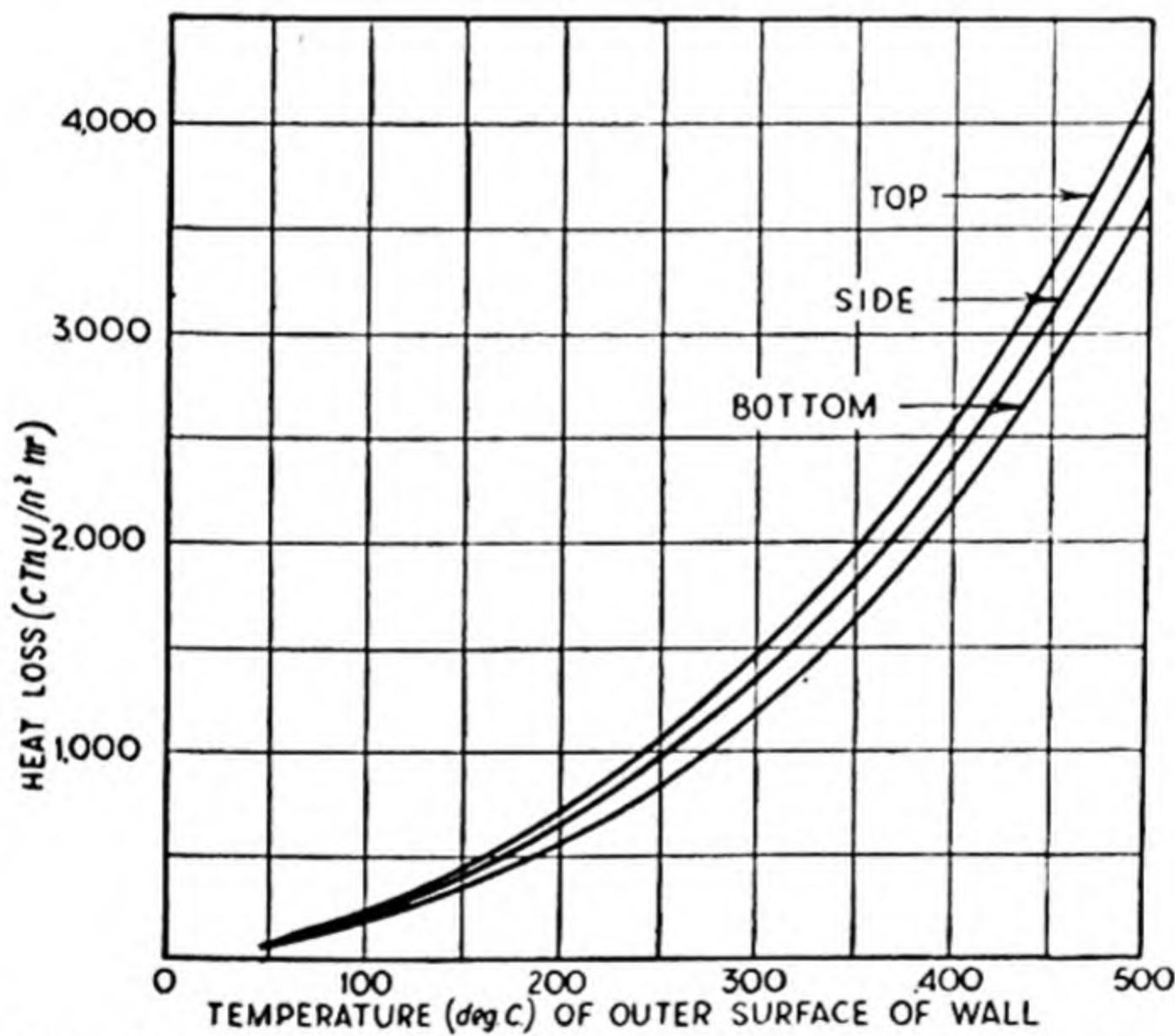


Fig. (4)17. Heat Loss from Furnace Walls, Bare Brick.

through the brick walls is by measuring the outside temperature of these walls at a number of evenly spaced points by means of a thermocouple such as that described in B.S.I. Code No. 1041.<sup>4.60</sup> From these measurements the heat loss per unit area of the furnace external surface can then be calculated from Fig. (4)16 and Fig.

<sup>4.60</sup> B.S.I. Code No. 1041, p. 56.



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(4)17. The curves of these figures have been worked out from the formula

$$H = C(T_4 - T_5)^{5/4} + 1.01 \times 10^{-8} E(T_{4a}^4 - T_{5a}^4), \text{ C.H.U./ft.}^2 \text{ hr.} \quad (4/22)$$

where  $C$  is the coefficient of natural convection for the type of surface concerned (formula 4/7) ;

$E$  is the emissivity ;

$T_4$  = outside wall temperature, ° C. ;

$T_5$  = atmospheric temperature (assumed in the case of these curves to be 18° C.) ; and

$T_{4a}, T_{5a}$  denote the corresponding temperatures in ° K. ( $T_a = T + 273$ ).

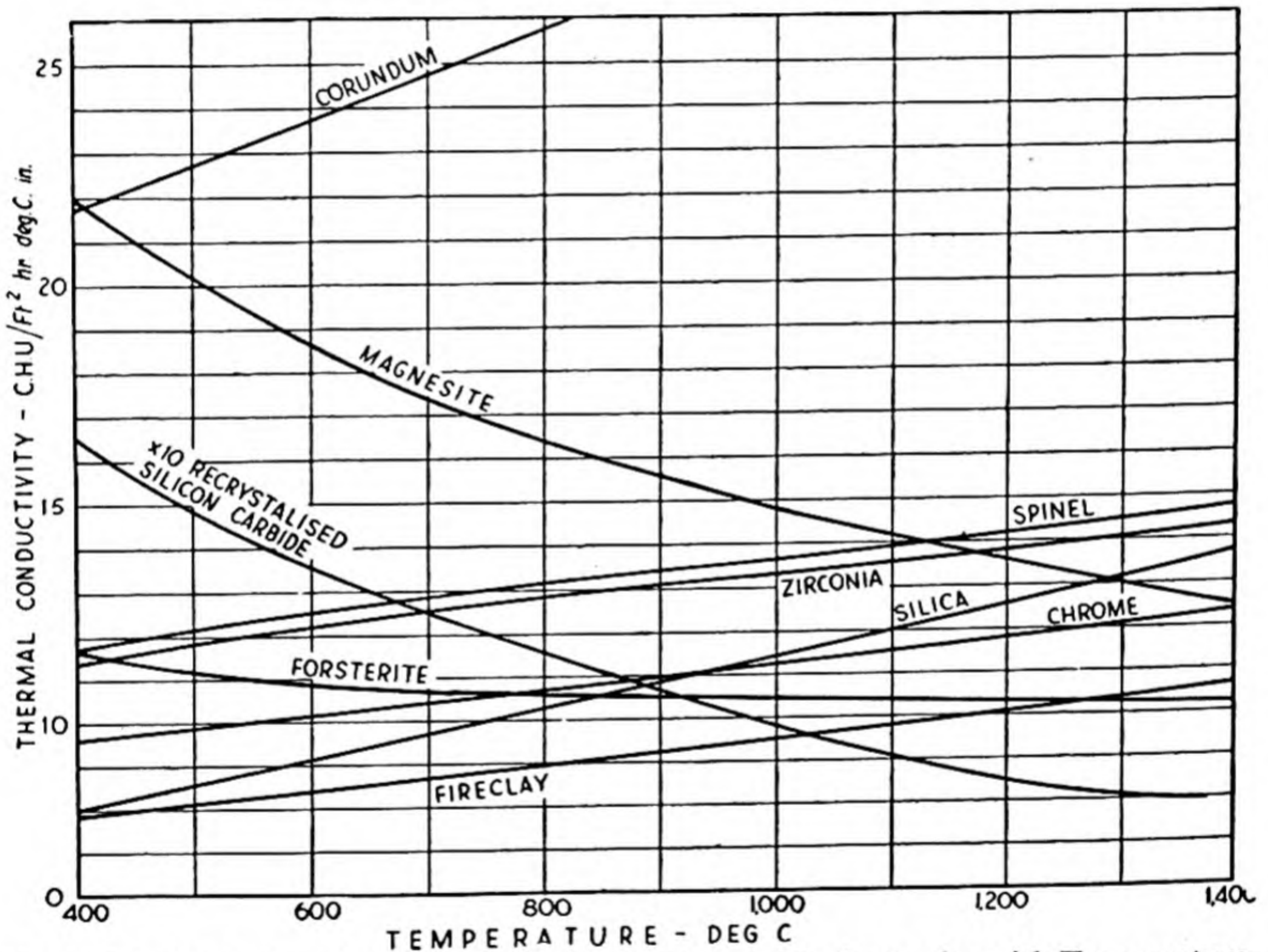


Fig. (4)18. Variation of the Thermal Conductivity of Refractories with Temperature.

The corresponding c.g.s. units are given also. Fig. (4)16 applies to the case of a furnace coated with metal plates painted with aluminium paint and thus having an emissivity  $E$  of 0.3. The three curves correspond respectively to the convection coefficient  $C$  of 0.21 (surfaces facing downwards, e.g. a fully ventilated hearth), 0.32 (vertical surface ; or mean values for top, sides and bottom of a complete rectangular furnace), and 0.41 (crowns). Fig. (4)17 corresponds to furnaces with bare brick on the outside so that  $E=0.9$ .



(2) *Estimation from measurement or guesswork of the inside wall temperature.* It is frequently possible to make a fairly reliable guess at the inside wall temperature of the furnace from a knowledge of the desired conditions for the stock. Alternatively, it may be possible to measure this temperature on an existing furnace by

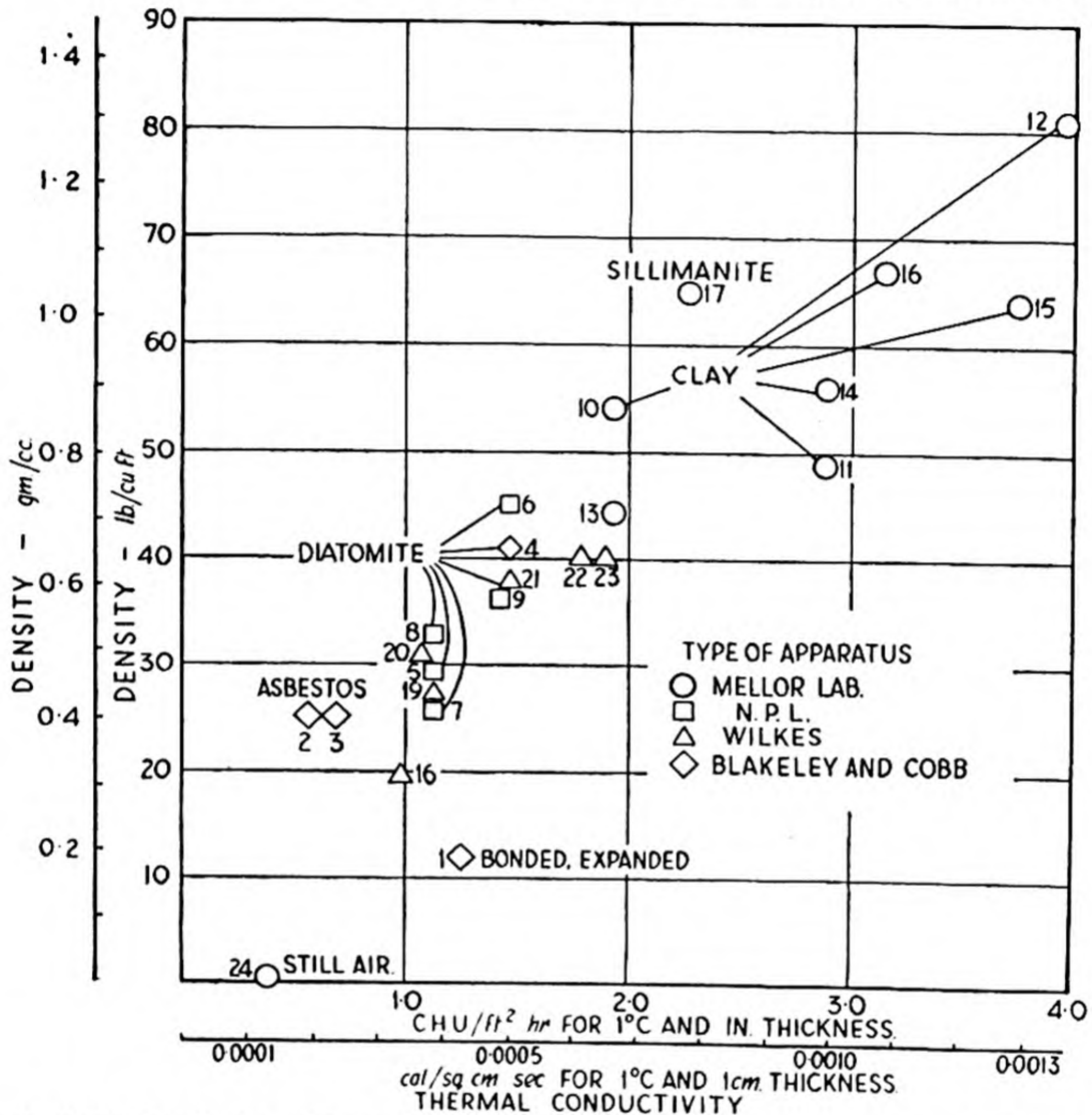


Fig. (4)19. Variation of the Thermal Conductivity of Insulating Bricks with Density. (Figures supplied by the British Ceramic Research Association.)

means of an optical pyrometer. In either case, from this temperature it is possible to make a rough calculation of the heat transfer through the wall using in addition the thickness of the various bricks of the wall and their thermal conductivities. For preference, the latter should be obtained from the makers of the particular bricks concerned, but if these are not available Fig (4)18 shows the variation of thermal conductivity of a number of commercial bricks with temperature, while Fig. (4)19 shows the thermal



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conductivity of a number of insulating bricks, plotted against the density. For convenience, tables of thermal conductivity have been calculated from these graphs, with the following results.

TABLE 4.9  
VARIATION OF THERMAL CONDUCTIVITY OF REFRACTORIES WITH TEMPERATURE  
(Units = C.H.U. in/ft.<sup>2</sup> hr. ° C.)

Temp. ° C.	Fire-clay	Silica	Chrome	Zir- conia	Spinel	Corun- dum	For- sterite	Recry- stallized silicon carbide (× 10)	Mag- nesite
400	7.8	8.0	9.6	11.4	11.7	21.7	11.6	16.5	22.0
600	8.4	9.1	10.2	12.2	12.5	23.7	10.9	13.6	18.6
800	9.0	10.3	10.7	12.8	13.1	25.7	10.5	11.6	16.4
1000	9.5	11.4	11.2	13.3	13.6	—	10.4	9.8	14.8
1200	10.0	12.6	11.8	13.8	14.2	—	10.3	8.5	13.6
1400	10.6	13.7	12.4	14.3	14.7	—	10.2	7.9	12.5

TABLE 4.10  
HIGH-TEMPERATURE INSULATORS AND DIATOMACEOUS INSULATORS

Temperature ° C.	20 lb./ft. <sup>3</sup>	30 lb./ft. <sup>3</sup>	40 lb./ft. <sup>3</sup>	50 lb./ft. <sup>3</sup>	60 lb./ft. <sup>3</sup>
200	0.47	0.75	0.95	1.2	1.4
400	0.55	0.85	1.08	1.4	1.65
600	0.65	0.96	1.23	1.6	1.93
800	0.72	1.06	1.43	1.8	2.2
1000	0.80	1.17	1.65	2.05	2.6

The thermal resistance  $R$  of a plane section of wall is then defined as  $d/k$ , where  $d$  is the thickness of the section in inches and  $k$  is the thermal conductivity of the brickwork taken from Table 4.9 at the estimated mean temperature of this section. It is usually possible to make a sufficiently close guess at this mean temperature. Thus, for example, if a single refractory is used without insulation the mean refractory temperature can be assumed to be the mean of the inner face temperature and the air temperature. If, on the other hand, one wants the mean temperature of a well-insulated inner refractory it can be assumed to be at the inner face temperature. If necessary, however, the mean temperature can be corrected by the method of successive approximation, that is, by inserting the values of  $k$  corresponding to the mean temperature derived from a first calculation into a second one. Then the



general equations for conduction of heat  $H$  through a composite brick wall consisting of three solid substances in series followed by heat transfer by radiation and convection from the final surface to surroundings at  $18^\circ \text{C}$ . is :

$$H = \frac{T_1 - T_2}{R_1} = \frac{T_2 - T_3}{R_2} = \frac{T_3 - T_4}{R_3} = f(T_4) \quad . \quad . \quad (4/23)$$

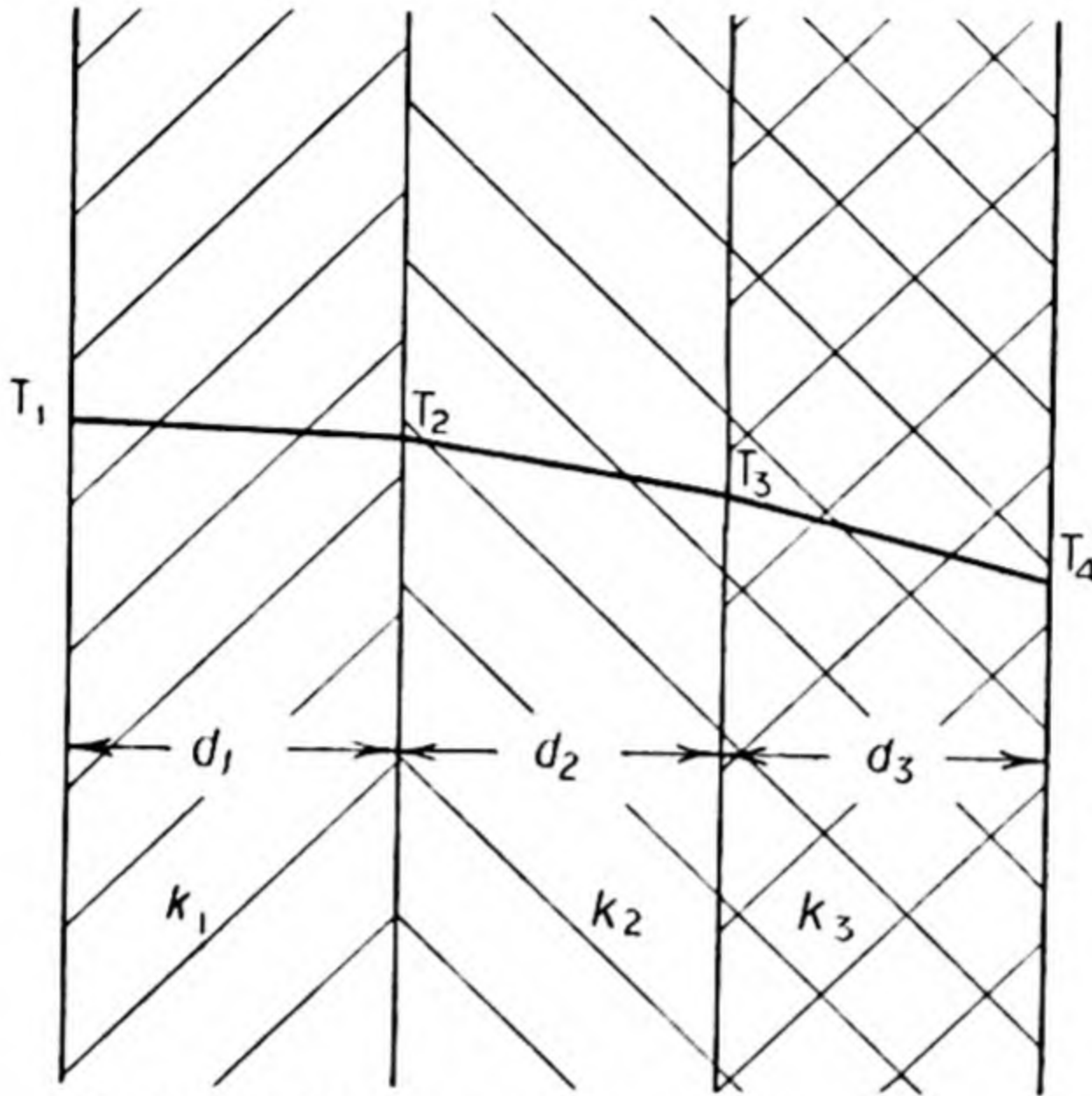


Fig. (4)20. Diagram of Heat Flow through a Wall.

where  $T_1$  = inside wall temperature  $^\circ \text{C}$ . ;

$T_2$  = temperature at first interface ;

$T_3$  = temperature at second interface ;

$T_4$  = outside wall temperature ; and

$R_1, R_2, R_3$  are the thermal resistances of the three wall sections.

$f(T_4)$  is the same function as in Case 1 above (formula 4/22), shown in Figs. (4)16 and (4)17, which connects the heat loss from the walls with their external surface temperature. This is shown diagrammatically in Fig. (4)20. The first three of these equations (4/23) are simply a restatement of the general law of thermal conduction, while the last expresses the convection and radiation from the outside of the wall as a function of  $T_4$ . To solve these equations  $T_2$  and  $T_3$  are eliminated to give

$$H = \frac{T_1 - T_4}{R_1 + R_2 + R_3} = f(T_4) \quad . \quad . \quad . \quad (4/24)$$



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This equation can then be solved graphically by finding the values of  $H$  and  $T_4$  corresponding to the point where the curve  $H=f(T_4)$  intersects a line through  $T_1$  having a slope  $\frac{1}{R_1+R_2+R_3}$ . By making this graphical solution for a large number of cases the curves of Figs. (4)21 and (4)22 have been calculated. To make use of these

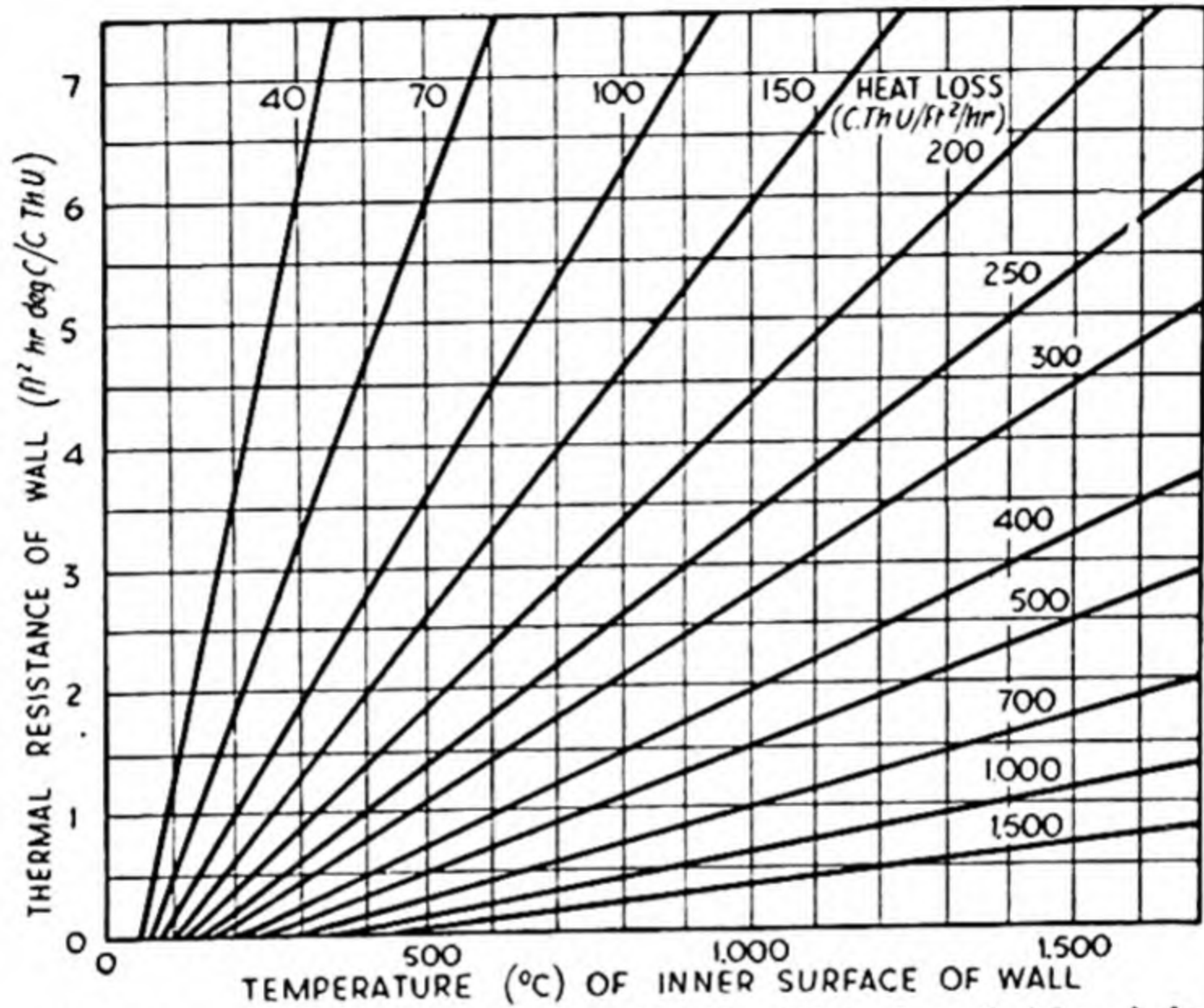


Fig. (4)21. Heat Loss Through Furnace Walls, Plated and Aluminium Painted.

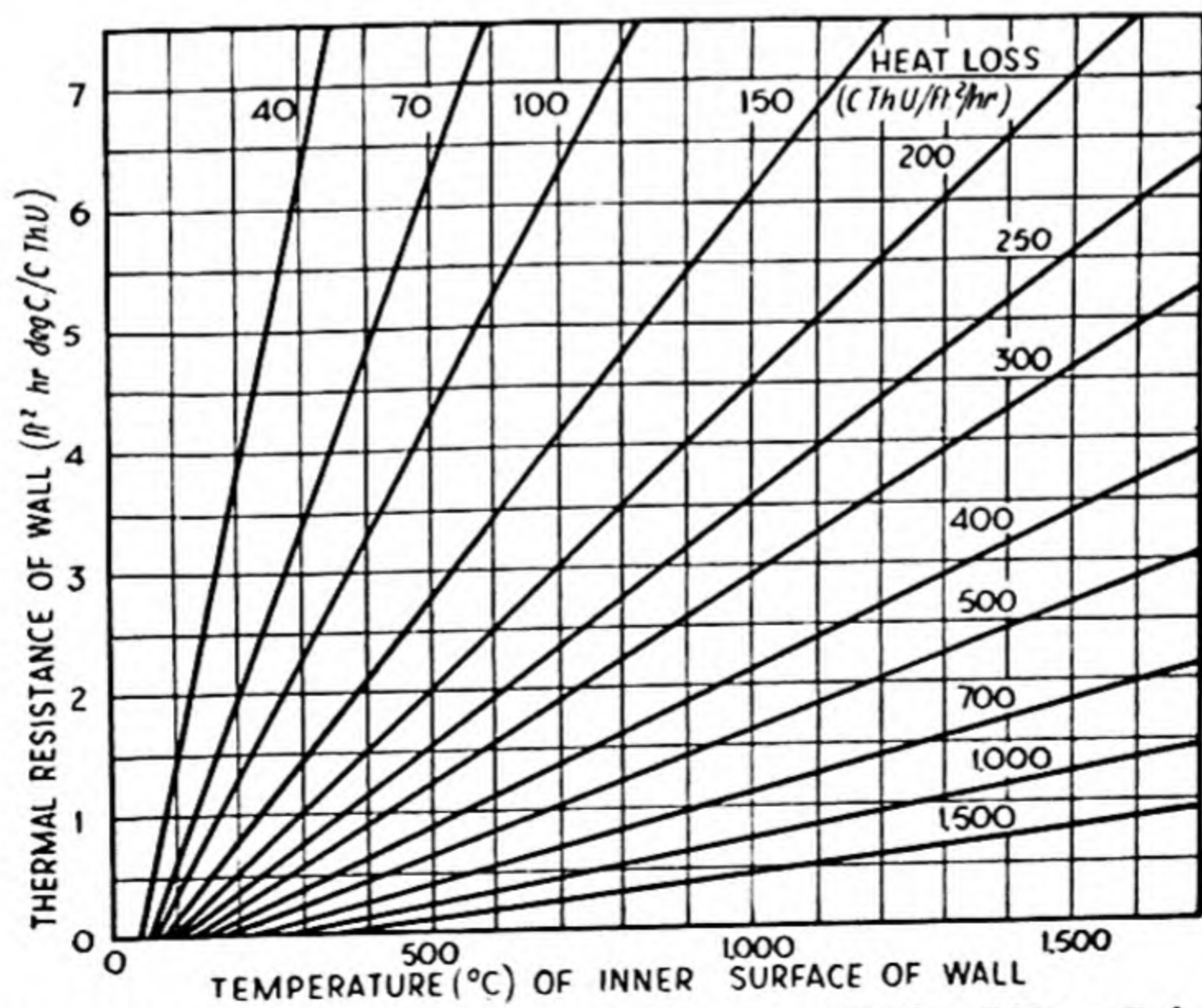


Fig. (4)22. Heat Loss Through Furnace Walls of Bare Brick.



figures the combined thermal resistance ( $R_1 + R_2 + R_3$ ) of the materials of the wall is calculated by adding the resistances of each material calculated from Table 4.9 as described above. From the value of  $R$ , the known inside wall temperature and the type of surface, the values of  $H$  and of the external surface temperature of the wall can then be read from Figs. (4)21 and (4)22.

From the calculated heat transfer per unit area ( $H$ ) the interface brick temperatures can be calculated from the formula

$$\begin{aligned} T_2 &= T_1 - HR_1 \\ T_3 &= T_1 - H(R_1 + R_2) \\ T_4 &= T_1 - H(R_1 + R_2 + R_3) \quad . \quad . \quad . \quad (4/25) \end{aligned}$$

where  $R_1$ ,  $R_2$  and  $R_3$  are the thermal resistances of the materials of successive layers of the wall. It is then possible to calculate more accurately the mean temperature of each layer, but it is rarely necessary to repeat the calculation using the new values, as the thermal conductivity of refractories does not vary sufficiently with temperature to bring the results outside the possible accuracy. The total heat loss from the furnace can be calculated by multiplying the value of  $H$  calculated for each surface by the appropriate area, which is best taken as the mean of the internal and external areas of that surface.

This method of calculation can also be used in cases where it is desired to design a brick wall with a certain insulating power subject to the condition that the inner face temperatures of one or more outer layers must not exceed certain values. An example of such a calculation is as follows :

It is required to build a furnace wall of a minimum heat capacity and good insulating properties to stand a hot-face temperature of  $1600^\circ \text{C}$ . The bricks available are :

- (1) A sillimanite brick capable of withstanding  $1600^\circ \text{C}$ . whose thermal resistance may be assumed to be equal to that of the firebrick of Table 4.9.
- (2) A refractory insulating brick (porous firebrick) capable of withstanding  $1350^\circ \text{C}$ . with a thermal conductivity of 1.5 C.H.U. in./ft.<sup>2</sup>  $^\circ \text{C}$ . hr. and density of 55 lb./ft.<sup>3</sup>
- (3) A diatomaceous earth insulating brick capable of withstanding  $900^\circ \text{C}$ . with a thermal conductivity of 0.85 C.H.U. in./ft.<sup>2</sup>  $^\circ \text{C}$ . hr. and density of 35 lb. ft.<sup>3</sup>

The problem is to calculate the thickness of the various bricks.



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In order to build a wall with low heat capacity and good insulation it is necessary to use as big a proportion as possible of the lightest brick and hence to have just enough of bricks No. 1 and 2 to bring the temperature down to  $900^{\circ}\text{C}$ . Similarly there should be just enough of brick No. 1 to bring the temperature down to  $1360^{\circ}\text{C}$ .

This is a slightly different problem from the one dealt with above, as none of the brick thicknesses are given, whereas the interface temperatures are fixed. A slightly different method of solution is accordingly necessary. A proposed method is as follows :

(1) Calculate the greatest permissible thickness ( $X$ ) of sillimanite brick consistent with the desired heat capacity, according to the figures given in Table 6.1, or decide a suitable thickness by experience. Then the heat loss in the ideal case is given by :

$$H = \frac{1600 - 1350}{X \cdot 0.11} \text{ C.H.U./ft.}^2 \text{ hr.}$$

(0.11 = thermal resistance of firebrick from Table 4.9).

(2) The necessary thickness ( $y$  in.) of brick No. 2 can then be calculated from the formula :

$$\begin{aligned} y &= \frac{1350 - 900}{H \cdot 1/1.5} \text{ inches} \\ &= \frac{1350 - 900}{1600 - 1350} \cdot \frac{X \cdot 0.11}{1/1.5} \text{ inches} \end{aligned}$$

(3) The necessary thickness ( $z$  in.) of brick No. 3 can then be calculated as follows :

Read off the value of  $R$  from Fig. (4)16 or (4)17 (whichever is appropriate), corresponding to the calculated value of  $H$  on the curve for  $T = 900^{\circ}\text{C}$ . Then  $z = \frac{R/1}{0.85}$  inches.

If the thicknesses  $y$  and  $z$  are not exactly standard brick sizes, it is necessary to choose the sizes immediately less than the calculated values. In this case  $H$  and the interface temperatures should be recalculated with the sizes finally selected, using the method described above.

Keller<sup>4.15</sup> has given a method of calculating the heat loss from a completely unventilated furnace hearth, i.e. one resting solid on the

<sup>4.15</sup> KELLER, J. D.: "The Flow of Heat Through Furnace Hearths," *Trans. A.S.M.E.*, 50, Paper F.S.P. 50-37 (1928).



ground. He assumes that the conductivity of the furnace hearth and of the ground is the same and is independent of temperature, and that the surface of the ground is at atmospheric temperature. The thermal conductivity taken by him for the ground and firebrick was 8.0 C.H.U. in./° C. ft.<sup>2</sup> hr. In order to calculate the total heat loss from a rectangular hearth of dimension  $a$  ft.  $\times$   $b$  ft. (where  $b$  is the shorter side), the equivalent thickness  $L_0$  of the hearth must be calculated. To do this first calculate  $L$  (the equivalent thickness excluding insulation) by multiplying by  $b$  the value of  $L/b$  which corresponds in Table 4.11 to the given value of  $a/b$ .  $L_0$  is the sum

TABLE 4.11

EQUIVALENT HEARTH THICKNESS (excluding insulation) DIVIDED BY HEARTH WIDTH AS A FUNCTION OF HEARTH SHAPE (from Keller)

$a/b$	$L/b$
1.0	0.23
2.0	0.25
3.0	0.255
4.0	0.26
5.0	0.26
6.0	0.265
$\infty$	0.27

of this value of  $L$  and  $R/1.50$ , where  $R$  is the thermal resistance of the insulating material at its estimated temperature calculated from its thickness and Table 4.9 as described above. The total heat loss through the hearth is then given by :

$$H = \frac{(T_1 - T_4) a \cdot b}{1.5 L_0} \text{ C.H.U./hr.}$$

$$\left( \text{for a circular hearth } H = \frac{(T_1 - T_4) \pi D^2}{1.5 L_0 \cdot 4} \text{ and } L = \frac{D}{4} \right)$$

In applying this formula two points should be especially noted :

- (i) Several months will elapse before the heat loss into the ground falls anywhere near as low as the final value.
- (ii) One-third of the heat loss takes place through the outer 5% of the hearth, hence variations in the method of applying the insulation at this point will result in great variations in the heat loss.

(3) *Calculation of the heat loss from the gas temperature and the brick resistances.* In certain cases where there is likely to be a large temperature difference between the gas and the inside of the



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brick it is not sufficiently accurate to make a guess as to the temperature of the inside surface of the brick ; but on the other hand it may be possible to calculate it from the convection conditions inside the system. In other words, it may be possible to calculate the first resistance  $R_0$  of the series connecting the hot gas with the atmosphere, which is that between the gas and the inside of the brick wall. This method has been worked out for the two cases of overhead and underground hot gas mains, e.g. those carrying

TABLE 4.12  
THERMAL RESISTANCE TO HEAT TRANSFER PER FOOT RUN BY  
CONVECTION FROM GAS TO TUBE WALLS  
(Reciprocal of heat transferred in C.H.U./ft. hr. °C. temperature difference)

Temp., °C.	Actual gas velocity in ft./sec.													
	2	3	4	5	6	7	8	9	10	11	12	15	20	25
200	0.94	0.68	0.54	0.45	0.39	0.35	0.31	0.28	0.26	0.24	0.22	0.18	0.15	0.12
300	1.1	0.83	0.66	0.55	0.48	0.42	0.38	0.35	0.32	0.29	0.27	0.22	0.18	0.15
400	1.2	0.90	0.72	0.60	0.52	0.46	0.41	0.37	0.34	0.32	0.30	0.25	0.20	0.17
500	1.3	0.93	0.74	0.62	0.53	0.47	0.43	0.39	0.35	0.34	0.31	0.26	0.21	0.17
600	1.4	1.0	0.82	0.69	0.59	0.52	0.47	0.43	0.39	0.36	0.34	0.28	0.23	0.19
700	1.5	1.1	0.85	0.72	0.62	0.55	0.49	0.45	0.41	0.38	0.35	0.29	0.23	0.19
800	1.5	1.1	0.89	0.74	0.64	0.57	0.51	0.46	0.43	0.39	0.37	0.31	0.25	0.21
900	1.6	1.2	0.94	0.79	0.68	0.60	0.54	0.49	0.45	0.42	0.39	0.33	0.26	0.22

The above figures are accurate to about 10%. They are calculated for a diameter of 4 ft. The error if  $D$  lies between 3 ft. and 5½ ft. is less than 10%. If  $D$  is outside this range multiply figures by  $(D/4)^{0.2}$ .

\* This table was calculated from the formula  $\frac{H}{\theta} = \left(\frac{k}{d}\right)^{0.2} \cdot (Vc)^{0.8} \cdot 100$

where  $\frac{H}{\theta}$  = heat transfer in C.H.U./ft<sup>2</sup> hr. °C.

$k$  = thermal conductivity in C.H.U./ft. sec. °C.

$d$  = diameter of main in feet.

$V$  = actual gas velocity in ft. per sec.

$c$  = specific heat of gas in C.H.U./ft.<sup>3</sup> °C. at constant pressure.

This formula is an expression for the mean of the curves plotted in Fig. 33 (p. 151) of *The Calculation of Heat Transmission*, by Fishenden and Saunders.

producer gas. In the case of overhead circular mains the method of calculation is exactly similar to that given above for the losses through furnace walls except that (1) the resistance  $R_0$  for the convection heat transfer from the gas to the inside wall must be added to those of the wall, and (2) allowance has to be made for the fact that the area changes from the inside of the main to the outside. This latter allowance is made by using the formulæ for conduction in concentric cylinders to calculate thermal resistances for unit length of main instead of unit area as used previously.



The thermal resistance ( $r_1$ ) to the heat transfer *per foot-run* by convection from the gas to the walls is obtained from Table 4.12 after the hot gas velocity ( $V$ , ft./sec.) has been calculated. Similarly the thermal resistance per foot run ( $r_2$ ) of the brick lining is obtained from Tables 4.13 to 4.15 according to the type of lining used (in the

TABLE 4.13

THERMAL RESISTANCE OF HOT GAS MAIN LININGS PER FOOT RUN  
Reciprocal of heat transferred in C.H.U./ft. hr. ° C. temperature difference for various internal and external radii of producer gas mains

*Low-conductivity Firebrick*

Internal dia., ft.	Internal radius inches	Thickness of walls in inches							
		3	4½	6	7½	9	12	18	24
2	12	0.067	0.112	0.155	0.190	0.225	—	—	—
2½	15	0.055	0.092	0.127	0.160	0.190	0.246	—	—
3	18	0.047	0.080	0.109	0.137	0.165	0.214	—	—
3½	21	0.040	0.069	0.096	0.122	0.145	0.190	0.268	—
4	24	0.0356	0.061	0.085	0.107	0.130	0.171	0.244	—
5	30	—	0.050	0.070	0.089	0.108	0.142	0.206	0.260
6	36	—	0.042	0.060	0.076	0.092	0.122	0.177	0.227
7½	45	—	0.034	0.048	0.062	0.075	0.100	0.146	0.190
9	54	—	0.029	0.041	0.052	0.064	0.085	0.126	0.164

It is assumed that the first inch is perfectly conducting, the remainder having a conductivity of 0.34 C.H.U./ft. hr. ° C. (0.0014 c.g.s. units).

TABLE 4.14

*High-conductivity Firebrick*

Internal dia., ft.	Internal radius inches	Thickness of walls in inches							
		3	4½	6	7½	9	12	18	24
2	12	0.034	0.055	0.078	0.095	0.112	—	—	—
2½	15	0.027	0.046	0.064	0.080	0.095	0.123	—	—
3	18	0.023	0.040	0.055	0.068	0.083	0.107	—	—
3½	21	0.020	0.035	0.048	0.061	0.073	0.095	0.134	—
4	24	0.018	0.031	0.043	0.053	0.065	0.086	0.122	—
5	30	—	0.025	0.035	0.045	0.054	0.071	0.103	0.130
6	36	—	0.021	0.030	0.038	0.046	0.061	0.089	0.114
7½	45	—	0.017	0.024	0.031	0.038	0.050	0.073	0.095
9	54	—	0.014	0.020	0.026	0.032	0.043	0.063	0.082

It is assumed that the first inch is perfectly conducting the remainder having a conductivity of 0.68 C.H.U./ft. hr. ° C.

case of unlined steel mains these are zero) and is added to that of the gas. The sum  $r_1 + r_2$  is then multiplied by the external circumference of the main in feet ( $\pi d_2$ ), where  $d_2$  is the outside diameter of the main, to give the total thermal resistance per square foot of outside area. The heat loss per square foot  $H$  is then obtained from Figs. (4)21 or (4)22 and hence the heat loss per foot-run  $h$  can be calculated from  $h = H \times \pi d_2$ .



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TABLE 4.15  
Half High-conductivity Firebrick, Half Insulating Brick

Internal dia., ft.	Internal radius, inches	Thickness of walls in inches							
		3	4½	6	7½	9	12	18	24
2	12	0.107	0.157	0.203	0.245	—	—	—	—
2½	15	0.089	0.132	0.171	0.206	0.239	—	—	—
3	18	0.075	0.113	0.147	0.179	0.205	0.261	—	—
3½	21	0.065	0.100	0.130	0.158	0.184	0.233	—	—
4	24	0.059	0.088	0.116	0.142	0.168	0.211	0.287	—
5	30	—	0.071	0.096	0.118	0.140	0.178	0.246	0.304
6	36	—	—	0.081	0.101	0.118	0.154	0.214	0.267
7½	45	—	0.050	0.066	0.083	0.098	0.128	0.180	0.226
9	54	—	0.042	0.056	0.070	0.083	0.109	0.153	0.197

It is assumed that the outer half of the thickness has a thermal conductivity of 0.73 C.H.U./ft. hr. °C., the first inch is perfectly conducting and the remainder has a thermal conductivity of 0.17 C.H.U./ft. hr. °C.: e.g. a 3-inch wall is taken to consist of

- 1 inch perfectly conducting;
- ½ inch of material with  $k=8.8$ ;
- 1½ inch of material with  $k=2.04$ .

Where none of Tables 4.13 to 4.15 correspond closely to the true conditions the correct resistance  $r$  can be calculated from the formula

$$r = \frac{1}{2\pi k_1} \log_e \frac{d_1}{d_2} + \frac{1}{2\pi k_2} \log_e \frac{d'}{d_2} = 0.366 \left( \frac{1}{k_1} \log_{10} \frac{d_1}{d'} + \frac{1}{k_2} \log_{10} \frac{d'}{d_2} \right)$$

The temperature drop of the gas per foot-run,  $dT/dx$ , is then calculated from whichever of the following formulæ is the most convenient.

(a) When the actual mean gas velocity ( $V_a$  ft./sec.) is known, then

$$\frac{dT}{dx} = \frac{h}{V_a \cdot A \cdot 3600 \cdot C_p} \text{ °C./ft.} \quad . \quad . \quad (4/26)(i)$$

where  $A$  = internal cross-sectional area in sq. ft.

$C_p$  = specific heat at constant pressure per unit volume measured at the actual gas temperature.

(b) When the N.T.P. gas velocity ( $V_o$  ft./sec.) is known,

$$\frac{dT}{dx} = \frac{h}{V_o \cdot A \cdot 3600 \cdot C_{po}} \text{ °C./ft.} \quad . \quad . \quad (4/26)(ii)$$

where  $C_{po}$  is the specific heat at constant pressure and at the actual temperature per unit volume measured at N.T.P.

(c) When the N.T.P. gas volume ( $Q_o$  ft.<sup>3</sup>/hr.) is known,

$$\frac{dT}{dx} = \frac{h}{Q_o \cdot C_{po}} \text{ °C./ft.} \quad . \quad . \quad (4/26)(iii)$$



(d) When the gas weight ( $W$  lb./hr.) is known,

$$\frac{dT}{dx} = \frac{h}{W \cdot C_p} \text{ } ^\circ \text{C./ft.} \quad . \quad . \quad . \quad (4/26)(iv)$$

where  $C_p$  is the specific heat at constant pressure and at the actual temperature per unit weight.

The temperature of the outside of the main can also be obtained from Fig. (4)16 or Fig. (4)17, while that of the inside of the brickwork is given by

$$T_1 = T - h \cdot r_1$$

and that of a brick interface (if any) by

$$T_2 = T - h(r_1 + r_2)$$

Underground mains can be treated in a similar way, except that the mechanism of heat transfer outside the brickwork is different from that outside overhead mains with exposed walls. Petri<sup>4.61</sup> measured the heat loss from buried steam pipes and concluded that they were the same as if the pipe were surrounded by a cylindrical layer of earth of thickness 8 ft. the outer surface of which was at atmospheric temperature. His experiments relate only to small circular pipes, but since his results show the general effect of the earth as an insulator they can probably be extrapolated to larger mains, especially as he studied the effect of insulation and found it to agree with the above-mentioned assumption as to the equivalent earth layer. The thermal conductivity of the earth was found to decrease very considerably with increase in temperature owing to the evaporation of moisture, and this is allowed for in his formula by a term which is exponential with temperature. He gives the formula

$$Q = \frac{2\pi}{\frac{\ln(d_2/d_1)}{\lambda_1} + \frac{\ln(d_3/d_2)}{0.25}} \{ (T_1 - T_3) + 343.5(0.9724^{T_2} - 0.9724^{T_3}) \} \quad (4/27)$$

where  $Q$  = heat loss per unit length in k.cal./m. hr. ;

$\ln$  = denotes logarithm to base  $e$  ;

$d_1$  = inner diameter of tube  
 $d_2$  = outer diameter of insulation } same units ;

$d_3$  = diameter of insulation equivalent thickness of earth layer ;

$A_1$  = thermal conductivity of insulation in k.cal./m. hr.  $^\circ \text{C.}$  ;

$T_1$  = inner temperature of tube,  $^\circ \text{C.}$  ;

$T_2$  = temperature of outside of insulation,  $^\circ \text{C.}$  ; and

$T_3$  = temperature of atmosphere,  $^\circ \text{C.}$

<sup>4.61</sup> PETRI : "Die Warmeverluste von Rohrleitungen in Erdreich," *Die Wärme*, p. 641 (1932).



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In the case of underground mains carrying hot gases the most important insulation is usually that due to the earth, and hence the outside of the brickwork will usually be above  $150^{\circ}\text{C.}$  ; in this case the term  $0.9724T_2$  can be neglected and the formula becomes

$$h = \frac{T_1 + 220}{r/2\pi + R_s} \text{ C.H.U./ft. hr. (assuming } T_3 = 14^{\circ}\text{C.) (4/28)}$$

where  $T_1$  = temperature of gas inside tube ;

$r = r_g + r_w$ , i.e. the sum of resistance of gas and walls calculated as for overhead mains (in case of non-circular mains the figures for circular mains of equal perimeter are used) ; and

$R_s$  = resistance of earth cylinder per unit length.

TABLE 4.16  
EQUIVALENT THERMAL RESISTANCE OF EARTH SURROUNDING  
BURIED PIPE

(Reciprocal of heat transferred in C.H.U./ft. hr.  $^{\circ}\text{C.}$ )

Outside diam. of brickwork in ft.	$R_s$
2	2.11
3	1.78
4	1.55
5	1.38
6	1.25
$7\frac{1}{2}$	1.10
9	0.99

In nearly all cases it will be found that  $R_s$  is much greater than  $r$ , so that  $r$  need not be calculated exactly. This means that the insulation of the dry earth round the main is much better than that of the brickwork, so that the latter will be very close to the gas temperature throughout.

**4.4.2. Other Causes of Heat Loss.** (1) *Heat loss through open doors.* In the case of openings large compared with the thickness of the wall the heat loss can be calculated as the product of the area of the opening and the black-body radiation at the furnace temperature, using Table 4.17 for the latter.

When the linear dimensions of the opening are not large compared to the thickness of the wall, the radiation loss is appreciably less than would be obtained by this method because the thickness of the wall screens off some of the radiation from reaching the outside. A factor by which the heat loss calculated on the un-



screened basis should be multiplied to allow for this effect is given by Hottel and Keller.<sup>4.62</sup> The factor rises from zero, when the wall is very thick compared to the size of the hole, past a value of 0.5–0.7 for holes the least dimension of which is equal to the thickness, and then rises steadily to unity for very thin walls. Details are given in Table 4.17.

TABLE 4.17  
“EMISSIVITY” OF FURNACE OPENINGS

Shape of opening	Ratio $\frac{\text{Diameter or least width}}{\text{Thickness of wall}}$							
	0.01	0.10	0.20	0.50	1	2	4	6
Circular	0.02	0.10	0.18	0.35	0.52	0.67	0.80	0.86
Square	0.02	0.11	0.20	0.36	0.53	0.69	0.82	0.67
Rectangular 2 : 1	0.03	0.13	0.24	0.43	0.60	0.75	0.86	0.90
Very long slot	0.05	0.22	0.34	0.54	0.68	0.81	0.89	0.92

(2) *Water cooling.* The heat lost to cooling water can be calculated very roughly by taking the area of water-cooled surface exposed in the furnace and multiplying this by the black-body radiation at the furnace temperature. Where the water-cooled

TABLE 4.18  
WATER COOLING LOSSES IN FURNACES (HEILIGENSTADT)  
*Units = C.H.U./hr. ft. length of pipe*

Ingot temperature, ° C.	Type of construction*		
	1	2	3
1,300	42,400	26,000	11,400
1,200	33,600	20,600	8,600
1,100	26,600	16,200	6,800
1,000	20,600	12,800	5,200
900	15,800	9,800	4,200
800	12,000	7,600	3,200
700	9,000	5,800	2,400
600	6,800	4,200	1,800
500	5,000	3,000	1,400
400	3,600	2,000	960
300	2,400	1,400	600
200	1,400	600	400

- \* 1. Fully exposed water-cooled skids.  
2. Skids protected by brick on the underside for half their length.  
3. Skids fully protected by brick on the underside.

<sup>4.62</sup> HOTTEL, H. C., and KELLER, J. D.: *Trans. A.S.M.E.*, p. 39 (1933).



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tubes are covered with brick, a better estimate will be obtained by calculating the heat transfer from the formula

$$H = K \frac{T_1 - T_2}{d} \text{ C.H.U./ft.}^2 \text{ hr.} \quad . \quad . \quad . \quad (4/29)$$

where  $H$  is the heat transfer per unit area of the pipe ;

$T_1$  is the furnace temperature ;

$T_2$  the water temperature ;

$K$  the thermal conductivity of the brick (Table 4/9) ; and

$d$  the thickness in inches.

Heiligenstadt<sup>4.49</sup> gives the figures of Table 4.18 for the heat lost to water-cooled skid pipes in pusher furnaces.

(3) *Air infiltration and gas leakage through the furnace brickwork.* If the furnace is at a pressure lower than that of the surrounding atmosphere an appreciable amount of air can be drawn in through the brickwork unless this is metal-cased. This "false air" will cause a lowering of the virtue of the combustion gases which will result in a greater quantity of heat being carried out of the furnace by these gases. In this case the only reliable way of assessing the results of such leakage is by measuring of the excess air in the gas leaving the furnace and comparing it with the excess of air calculated from measurement of that entering by the proper route. Where the leakage is early on in the air flow circuit, e.g. into the ingoing checkers, its effect can be offset to a large extent by reducing the amount of air going in by normal means. No simple formulæ can be given for calculating air inleakage, since it depends too much on the state of the brickwork and on the number of holes and gaps in the furnace structure, which probably explains why some authors maintain that it is a very small effect, while others say that it results in the heat losses being approximately 20 or 30% greater than those calculated by the methods given in 4.4.1.

Maase<sup>4.63</sup> gives the following formula for the heat flow through brickwork due to gas leakage.

$$H_R = V \cdot c_p (T_1 - T_a) \text{ C.H.U./ft.}^2 \text{ hr.}$$

where  $V$  is the volume flowing through  $n$  ft.<sup>3</sup>/hr.-ft.<sup>2</sup> of wall.

$C_p$  = specific heat of gas per ft.<sup>3</sup> ;

$T_1$  = furnace temperature, ° C. ;

$T_a$  = atmospheric temperature, ° C. ; and

4.49 HEILIGENSTADT: *Warmetechnische Rechnungen für Bau u. Betrieb von Öfen* (Düsseldorf, 1925).

4.63 MAASE: *Feuerfest*, p. 97 (1932).



$V$  is given by

$$V = \epsilon h / d \text{ } n \text{ ft.}^3/\text{ft.}^2 \text{ hr.}$$

where  $h$  = pressure difference in inches w.g. ;

$d$  = thickness of wall in inches ; and

$\epsilon$  varies between 33 for very carefully laid walls and 660 for walls with fine cracks—the value being usually around 350–500.

Köfler<sup>4.64</sup> quotes a case where 28% of the fuel furnace was saved by making it gas-tight with asphalt.

(4) *Heat losses through projecting parts of the stock.* This is of importance only in the case of the reheating of large steel ingots, where the conduction out of the ingot may be very large. It should be noted that this form of heat loss is particularly undesirable because it implies not only a loss of heat from the furnace gases but actually from that part of their heat which has been usefully transferred to the charge. Trinks suggests that as a rough working approximation the stub should be replaced by an equivalent opening from which heat can radiate without obstruction, this opening being 15% of the cross-section of the projecting part of the ingot during the heating period, and 30% during the steady state period. A more accurate solution can be obtained by calculating the heat conduction by the use of equation 4/4 ; methods of doing this are described in detail in *Heat Conduction* by Ingersoll, Zobel and Ingersoll (McGraw Hill), Chap. 11 ; a convenient graphical method was developed by Awbery and Schofield.<sup>4.16</sup>

## 4.5. Heat Flow in the Unsteady State

**4.5.1. General Methods.** There are three cases in which the flow of heat varies with time (i.e. where the heat flow causes a continual change in the temperature field) which are of practical importance in furnace design. These are :

- (1) The calculation of the time taken to heat up material in a furnace, e.g. in a batch-type billet-heating furnace and in an open-hearth furnace during the melting period. (This is discussed in 4.5.2.)

<sup>4.64</sup> KÖFLER : *Archiv. f. d. Eisenhüttenwesen*, p. 493 (1932).

<sup>4.16</sup> AWBERY, J. H., and SCHOFIELD, F. H.: *Proc. Intern. Conf. Refrig.*, 5th Congr., 3, p. 591 (1929).



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- (2) The heat flow into and out of furnace walls during the heating up and cooling down of the furnace (4.5.3).
- (3) The calculation of the heat transfer between gas and bricks in a regenerator (4.5.4).

In all cases of unsteady heat transfer the basic differential equation governing the temperature of a small element of volume of the heated body is that given in Chapter 4.2.1 under the paragraph on conduction, namely :

$$K \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = C \frac{\partial T}{\partial t} \quad . \quad . \quad . \quad (4/4)$$

It will be seen from this equation that the only property of the materials which is of importance in deciding the relation between temperature and time at any point is the ratio of  $K$ , the thermal conductivity, and  $C$ , the specific heat per unit volume. This ratio is known as the diffusivity and values of it for various materials are given in Table 4.19.

TABLE 4.19  
THERMAL DIFFUSIVITY OF REFRACTORY BRICKS  
*Units = ft<sup>2</sup>/hr.*

Temperature °C.	Silica	Firebrick	Semi-silica
600	0.0140	0.0100	0.0100
700	0.0144	0.0100	0.0112
800	0.0152	0.0104	0.0120
900	0.0156	0.0102	0.0128
1000	0.0164	0.0112	0.0140
1100	0.0172	0.0120	0.0152
1200	0.0184	0.0128	0.0164
1300	0.0196	0.0136	0.0176

There are three methods of estimating unsteady state-heat flow, namely,

- (1) the complete calculation from the differential equation, using the known boundary conditions for the surface temperature of the body\* ;
- (2) the use of approximate graphical methods based on the differential equation ; and

\* The mathematical processes necessary for these complete calculations are very fully discussed by Carslaw and Jaeger.<sup>4.65</sup>

<sup>4.65</sup> CARSLAW and JAEGER : *Conduction of Heat in Solids* (Oxford, 1947).



- (3) the application by similarity methods (or by crude empirical formula) of results obtained with models or other comparable furnaces.

The first two methods are open to the serious objection that it is always necessary to make simplifying approximations before the equations can be solved ; thus for example, in calculating the heating of a steel billet it is necessary to assume either that the surface is instantaneously raised to a known temperature or at least that it is instantaneously submitted to radiation from a black-body at a known temperature and has a known emissivity. It is also usual to assume that the specific heat is either constant or a simple function of temperature, i.e. no allowance is made for the abnormalities which occur at certain temperatures. Where such allowance is made the mathematical solution of the equation becomes almost impossible, although specific cases can be solved by integrating machines<sup>4.17</sup> or the method of finite differences. For this reason calculations by the first method are necessarily subject to considerable doubt and this is why the rough approximations to mathematical accuracy of the second method give usually as accurate an answer as the complete calculation. (The second method of course requires the same physical approximations as the first.) The first method, however, is of great value in that it can be used to give a quantitative account of the effect of the various variables on the heat transfer rates and rates of rise of temperature. Examples of such quantitative formulæ are as follows :

(1) The rate of heat penetration into a semi-infinite solid with one plane face instantaneously raised to a certain temperature and then maintained at that temperature is such that the same intermediate temperature is reached at two distances  $x_1$  and  $x_2$  from the face in times  $t_1$  and  $t_2$  respectively given by the relation  $t_1/t_2 = x_1^2/x_2^2$ : in other words, the time taken to reach a given temperature is dependent not on the distance from the surfaces but on the square of the distance. This is a particular application of the fact that the linear dimensions  $L$ , the diffusivity  $K/C$  and the time  $t$  can be grouped together in the dimensionless group  $CK^2/Kt$ . Hence by the use of similarity considerations or the use of general theory, for a given shape and surface temperature distribution it can be shown that the

<sup>4.17</sup> JACKSON, R., SARJANT, R. J., WAGSTAFF, J. B., EYRES, N. R., HARTREE, D. R., and INGHAM, J.: "Variable Heat Flow in Steel," *J.I.S.I.* (July 1944).



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internal temperature distribution will depend only on this group  $CL^2/Kt$ .

(2) Similarly the flow of heat into unit area of semi-infinite solid at time  $t$  is given by

$$H = \frac{K(T_1 - T_0)}{12\sqrt{Kt/C}} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (4/30)$$

where  $T_1$  = temperature to which surface is suddenly raised ; and  $T_0$  = initial temperature of the whole semi-infinite solid.

In this way the basic laws of heat transfer can be used to show the effect of changes of the dimensions of the material being heated, of the thermal diffusivity or of the rate of heat supply. Where calculating machines such as the Bush differential analyser or numerical methods are used to solve the differential equations the result is not in the form of general formulæ but only of specific solutions, so that these machines can only be used to find the effect of specific variables by calculating a number of special cases.

The *graphical method* of Schmidt<sup>4.66</sup> is fully discussed in Fishenden and Saunders (pp. 77–83) and can be used in cases where it is desired to know the approximate temperature distribution in a solid with appropriate simplifying assumptions and without the complication of a full mathematical solution.

It is based on the fact that, if there is linear conduction of heat along the line connecting three equidistant points,  $A, B, C$  ( $AB = CD = x$ ), then point  $B$  has at any given moment a temperature equal to the mean of those temperatures which points 1 and 3 had at a time  $t_0 = cx^2/2k$  earlier. Hence if a line is drawn on a  $T-x$  diagram connecting the temperature of the points  $A$  and  $C$  at one time  $t$  its midpoint will indicate the temperature of point  $B$  at a time  $t + t_0$ . A series of such lines can be drawn for successive times once  $t_0$  has been calculated by dividing the whole block of material into a number of equal steps  $x$  and treating each three successive points as  $A, B$  and  $C$ .

Southwell's relaxation method has been applied to the numerical solution of heat transfer problems of both steady and unsteady state by Emmons<sup>4.67</sup> and Crank and Nicholson<sup>4.68</sup> ; this may also be classified under the heading of approximate solutions of the

4.66 SCHMIDT : *Föpples Festschrift*, p. 179 (1924).

4.67 EMMONS, H. W. : "The Numerical Solution of Heat Conduction Problems," *Trans. A.S.M.E.*, **65**, p. 607 (1943).

4.68 CRANK and NICHOLSON : *Proc. Camb. Phil. Soc.*, **43**, p. 50 (1947).



differential equations. This is a successive approximation method, applying a series of operations in such a way that a trial solution is "tuned up" to a close approximation to the correct solution.

The third method of estimation, by extrapolation of experimental results, is considered in connection with a particular problem in the next section.

#### 4.5.2. The Heating Up of the Charge in a Batch-type Furnace.

Where a *thin* body is being heated in a batch-type furnace it may be assumed to be at a uniform temperature throughout and the only complexity of calculation arises from the fact that the rate of heat transfer falls off as the temperature of the body rises. In such a case the temperature  $T_c$  of the charge after a time  $t$  is given by

$$T_c = T_f - (T_f - T_o)e^{-(\alpha t/gC)} \quad . \quad . \quad . \quad (4/31)$$

where  $T_f$  is the effective furnace temperature (assumed constant) ;

$T_o$  is the temperature of the batch at the time of insertion ( $t=0$ ) ;

$\alpha$  is the heat transfer coefficient by radiation and convection (see Sections 4.3.2 and 4.3.3) ; C.H.U./ft.<sup>2</sup>—hr.—°C.

$g$  is the weight of the body in lb./ft.<sup>2</sup> of exposed surface ; and

$C$  is its specific heat (C.H.U./lb. °C.).

Conversely the time to reach a desired temperature  $T_c$  is given by

$$t = \frac{g \cdot C}{\alpha} \log_e \frac{T_f - T_c}{T_f - T_o} \quad . \quad . \quad . \quad (4/32)$$

When the body being heated is thick enough for the temperature gradient through it to be significant an extra complication arises. As far as the average temperature of the body is concerned the thickness may be treated merely as an increase in the time of heating resulting from the reduction of the overall value of  $\alpha$ . Heiligenstadt has calculated the factor  $r$  by which the term  $t$  in equations (4/31) and (4/32) must be divided to obtain the true time, by using Schmidt's graphical method. This factor is a function of

the dimensionless parameter  $\frac{\alpha}{K}d$ , where  $K$  is the thermal conductivity of the body and  $d$  its characteristic thickness, and varies between 1 for  $\frac{\alpha}{K}d=0$  (very thin sheets) and 0.4 for slabs for which  $\frac{\alpha}{K}d=4$ .

Thus in the latter case the actual time is increased  $2\frac{1}{2}$  times compared to that calculated from (4/32). In cases where the maximum furnace temperature is fixed by the fact that the temperature difference in



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the body must not exceed a certain value it is necessary to calculate the actual temperature distribution in the body.

Sarjant<sup>4.69</sup> has summarised the results of various workers on this subject. He gives curves for the time of heating which are nearly linear when plotted against the smallest dimension of round or square billets ; thus the time of heating billets of 16 inches diameter varies according to the furnace temperature or permissible rate of heating from 1200 to 300 minutes ; smaller size billets have times in direct proportion. Alternatively he gives a rule of 90 minutes for every 4 inches of diameter for mild steel. For 60-inch ingots the times vary between 36 and 43 hours. These results cannot, however, replace the estimation by one of three methods discussed in 4.5.1 since they do not relate the heating time to the temperature of the furnace.

The best example of the third method of estimation is that of Janitzky.<sup>4.70</sup> Janitzky measured the heating rates of small bars of known diameter ( $D$ ) in furnaces and derived the following equation for extrapolation of these results.

Time in hours required to heat the body

$$=K_1 \cdot K_2 \cdot K_3 \cdot D \quad . \quad . \quad . \quad . \quad . \quad (4/33)$$

where  $K_1$  has the value of 0.013 for spheres or cubes, 0.019 for round, square or octagonal billets, and 0.038 for plates ;

$K_2$  depends on the temperature and takes account of the variation of properties of the metal and its emissivity ; and

$K_3$  has the value 1.0 if the furnace is hot when the charge is inserted and 1.5 if the furnace is cold.

Since, however, the measurements were only carried out on one size of specimen the use of this formula is not to be recommended.

Recently this subject has been tackled<sup>4.17</sup> in a more fundamental manner by calculating the temperature distribution in the billet, making allowance for the variations of specific heat, using the Bush differential analyser and an electric machine for solving the

4.69 SARJANT, R. J. : *J. Inst. Fuel*, p. 368 (1937).

4.70 JANITZKY, E. J. : (1) "Characteristics of Air Cooling Curves," *Trans. Amer. Soc. Steel Treating*, 3, p. 335 (1932). (2) "Characteristics of Heating Curves, Their Applicability for Calculating the Time Required to reach Constant Head Temperature," *Trans. Amer. Soc. for Steel Treating*, 5, p. 201 (1924).

4.17 JACKSON *et al.* : "Variable Heat Flow in Steel," *J.I.S.I.* (1944).



equations. The results have been presented in the form of a nomograph from which the necessary "soaking time" to obtain a certain maximum temperature difference can be calculated if the time taken to heat up the ingot for the last 100° C. and the ingot properties are known.

**4.5.3. Furnace Walls.** There are two cases in which the calculation of the unsteady-state heat transfer in furnace walls is of importance.

(1) *Calculation of the time for the walls to reach any given temperature during initial or cyclic heating up of a furnace.* It may be desired to calculate this time for a given rate of heat input in order to know how soon a furnace can be put into operation, or else, as in the case of the heating of steel billets, it may be necessary to calculate how the rate of heating must be adjusted to give the desired rate of heating of the walls.\* In this case calculation is only a very rough guide because the whole of the flame properties vary during this period, and what is in fact done is to adjust the fuel supply to obtain the rate of heating which experience has shown to be most beneficial to the refractories.

In the special case where the flame is put on immediately to an empty furnace at the full rate that will be required to maintain the empty furnace at the final temperature, a very crude formula  $t = D^2/8$  has been used to calculate the time ( $t$ ) in hours to reach an approximately final state for an uninsulated firebrick wall of thickness  $D$  inches.<sup>4.59</sup>

(2) *Calculation of the heat losses, additional to those which occur during the steady state, due to cutting off the firing of the furnace, e.g. during the night or the weekend.* These are of importance for the complete heat balance. (Where the heating is left on but the furnace not charged, the heat losses can be calculated by the steady-state methods described in Section 4.4.)

The total loss of heat from the brickwork may be roughly estimated for the general case from one method  $A$  or, where the furnace is shut down for regular periods of 8 or 16 hours out of the 24, another method,  $B$ , given by Heiligenstadt may be used.

Method  $A$  is based on the fact : (a) at the moment the heat supply is cut off, the rate of heat loss per square foot is the same ( $H_o$ ) as it was when the furnace was running in the steady state ; and

\* In heating up an open-hearth furnace the rate of heating has to be adjusted very carefully when the crown is at the silica inversion temperature.  
<sup>4.59</sup> ETHERINGTON : *Modern Furnace Technology*, p. 203.



(b) that the heat loss after infinite time is equal to the total heat stored initially in the brickwork. The method depends on drawing a smooth curve for the total heat loss which has a tangent at  $t=0$  to give the initial rate  $H_0$  and comes asymptotically to the value equal to the total heat of unit area of the brickwork.

It is assumed that there is no flow of air through the inside of the furnace during idle periods so that the mechanism of heat loss is the same as that during steady-state periods, viz. conduction through the walls followed by convection and radiation.

The approximate method of calculation is as follows :

- (i) Calculate by the methods of 4.4 the heat loss from the furnace per unit area in the steady state. Let this be  $H_0$  C.H.U./ft.<sup>2</sup> hr. Calculate also the brick face temperature.
- (ii) Calculate the total heat  $Q_m$  stored in the brickwork per ft.<sup>2</sup> when in the steady state above atmospheric temperature (taken as 18° C.). This is given by

$$Q_m = cp \frac{d}{12} \left( \frac{T_1 + T_0}{2} - 18 \right) \text{C.H.U.} \quad . \quad . \quad . \quad (4/34)$$

where  $d$  = thickness of each section of brick in inches ;

$T_1$  = assumed internal temperature of brick in steady state, ° C. ;

$T_0$  = calculated external temperature of brick (4.4) in steady state, ° C. ;

$c$  = mean specific heat of wall at mean wall temperature (C.H.U./lb. ° C.) ; and

$p$  = density of brick, lb./ft.<sup>3</sup>

- (iii) Hence, calculate the time  $t_m$  in which all the heat stored in the brickwork would disappear if heat losses continued at the initial rate. This is given by

$$t_m = Q_m / H_0.$$

- (iv) If the time for which the heat is shut off is more than twice  $t$  then the total heat losses in the shut-off period may be taken as  $Q_m$ .

If the total shut-off period is less than one-third  $t_m$ , then the heat losses may be taken as continuing at the initial rate  $H_0$  throughout the period.



If  $t_m$  lies between these values then the total heat lost in the shut-off period can be roughly calculated from the formula

$$Q = Q_m(1 - e^{-t_o/t_m}) \text{ C.H.U./ft.}^2 \quad . \quad . \quad . \quad (4/35)$$

where  $t_o$  is the total shut-off period.

Values of the function in the bracket are given below :

$t_o/t_m$	$(1 - e^{-t_o/t_m})$
0.50	0.39
0.75	0.53
1.00	0.63
1.25	0.71
1.50	0.75
1.75	0.83
2.00	0.86

As an example of such a heat loss expressed in terms of the total additional fuel consumption necessary due to shutting down, the observations of Leckie<sup>4.71</sup> may be cited. Leckie found that in a typical open-hearth furnace the melting time was about  $1\frac{1}{2}$  hours longer for the first melt of the week than for the best melt of the week, and the thermal efficiency was reduced from the normal value of 25% to 20% for the first melt.

Heiligenstadt's method (B) is based on a detailed calculation by the Schmidt process of the temperature distribution in the wall, assuming this to be of uniform composition and having the properties of normal firebrick, for two cases of daily intermittent operation. He gives the following table for the heat lost by external wall losses, expressed as a percentage of the total heat  $Q_m$  stored in the wall in terms of the type of daily cycle and the percentage cooling of the inside face.

TABLE 4.20  
FRACTION OF HEAT STORED IN WALLS LOST DURING  
DAILY CYCLIC SHUTDOWNS (i.e. Heiligenstadt)

Working time, hours/day.	$Q/Q_m \times 100$		
	Cooling of inside face as percentage of working temperature.		
	80%	60%	40%
16	96	93	90
8	91	87	79

4.71 LECKIE, A. H. : "The Study of the Thermal Performance of Open-Hearth Furnaces by the Correlation of Operating Data," *J.I.S.I.*, p. 419 (Jan. 1944).



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If all the cooling takes place from the outside, then the drop of temperature of the inside face required for Table 4.20 can be obtained from his figures given in Table 4.21.

TABLE 4.21  
FINAL TEMPERATURE OF INSIDE FACE AS PERCENTAGE  
OF WORKING TEMPERATURE

Hours/day	Wall thickness—_inches		
	10	15	20
16	58%	69%	77%
8	39%	52%	62%

If the furnace which is being calculated is not built, then the temperature drop must be obtained from Table 4.21 and the result used in Table 4.20 to calculate the fractional heat loss. If the furnace already exists and is working, the temperature drop should be measured and used in Table 4.20. Comparison of the observed value with Table 4.21 will show how much heat is being lost by internal losses (cold air circulation inside the furnace).

The total increase in fuel consumption due to cooling down the furnace and then reheating it involves considerably more than the total loss of heat from the brickwork during the period of shutting down. This is because the amount of heat required by the furnace to bring it back to temperature is the heat given to the brickwork plus the additional heat carried out by the gases during this period. Hence, in order to calculate the loss due to a shutdown period, it is necessary to calculate the loss of heat concerned in the brickwork and divide it by an efficiency factor  $\eta$  which will be very roughly

given by  $\eta = \frac{T_i - \frac{1}{2}T_f}{T_i}$ , in the case of furnaces which have cooled

almost to atmospheric temperature, and by  $\eta = \frac{T_i - T_f}{T_i}$  in the case of slight cooling, being intermediate between these two values in other cases.

$T_i$  = theoretical flame temperature ; and

$T_f$  = furnace temperature in the steady state.

**4.5.4. Regenerators.** In spite of the fact that a regenerator is an example of the unsteady state, the methods of calculating the preheat temperature of the heated gases and the final temperature of the waste gases can be based upon those for the steady-state



recuperator (Section 4.3.5). The principle of this method of calculation is that the mean temperature of each gas during its half-cycle at any point in the system is taken as the appropriate temperature in an equivalent recuperator ; the overall coefficient of heat transfer  $\alpha$  is replaced by a coefficient  $\phi$  expressed in C.H.U./ft.<sup>2</sup> ° C.  $\phi$  is given by the equation :

$$\frac{1}{\phi} = \frac{1}{\alpha_1 t_1} + \frac{1}{\alpha_w t_0} + \frac{1}{\alpha_2 t_2} \quad . \quad . \quad . \quad (4/36)$$

where  $t_0$  is the complete reversal period (both directions of flow) in hours ;

$t_1$  and  $t_2$  are the times of the portions of the cycle corresponding to gas and air flow respectively ; and

$\alpha_1$  and  $\alpha_2$  are the corresponding gas-brick heat transfer coefficients in C.H.U./ft.<sup>2</sup> ° C. hr.

In this equation  $\alpha_w$  no longer corresponds to the coefficient for conduction through the wall, but is a factor expressing the heat soakage into and out of the brickwork ; it is described by Rummel<sup>4.72</sup> as the thermal hysteresis factor and has the approximate value

$$\alpha_w = 0.0051 \text{ C.H.U./ft.}^2 \text{ hr. } ^\circ \text{C.} \quad . \quad . \quad (4/37)$$

It is in the factor  $\alpha_w$  that account is made for the fact that the heat flow is unsteady. For brick thicknesses less than 2 inches  $\alpha_w$  depends also on the brick thickness, especially for high values of  $\alpha_1$  and  $\alpha_2$  and for long reversal periods ; this is because under these conditions the temperature at the centre of each brick begins to follow the reversals and its heat absorption capacity is limited by its volume instead of, as with thicker bricks, only by the resistance to heat flow into it. Values of  $\phi$  are given by Heiligenstadt<sup>4.49</sup> in tables on pp. 160-4 (he calls this factor  $\epsilon$ ) ; these tables can be used in cases where the brick thickness is smaller than 2 inches or the soaking time is very long.

$\alpha_1$  is the sum of the heat transfer coefficient  $\alpha_{1r}$  due to radiation from the waste gases and that due to convection  $\alpha_{1c}$ . According to Rummel

$$\alpha_{1c} = \frac{1.26 V_o^{1/2}}{d^{1/3}} \text{ C.H.U./hr. ft.}^2 \text{ } ^\circ \text{C.} \quad \text{for straight-through checkers ; and}$$

$$\alpha_{1c} = \frac{1.39 V^{1/2}}{d^{1/3}} \text{ C.H.U./hr. ft.}^2 \text{ } ^\circ \text{C.} \quad \text{for double-staggered checkers} \quad . \quad . \quad (4/38)$$

<sup>4.72</sup> RUMMEL : *J. Inst. F.*, p. 160 (1931).

<sup>4.49</sup> *Warmetechnische Rechnungen für Bau u. Betrieb von Ofen.*



#### 4.5.4 THE SCIENCE OF FLAMES AND FURNACES

where  $V_o$  = N.T.P. velocity of gas in free area of checkers ft./sec. and  
 $d$  = side of square flue in feet.

(where the cross-section is not square the hydraulic diameter is used for  $d$ ; this is equal to

$$\frac{4 \cdot \text{cross-sectional area of flue (ft.}^2\text{)}}{\text{perimeter of flue (ft.)}}$$

These relations are true at  $600^\circ \text{C.}$  : for every  $30^\circ \text{C.}$  above or below this temperature  $\alpha_{1c}$  should be increased or decreased by 1%. For

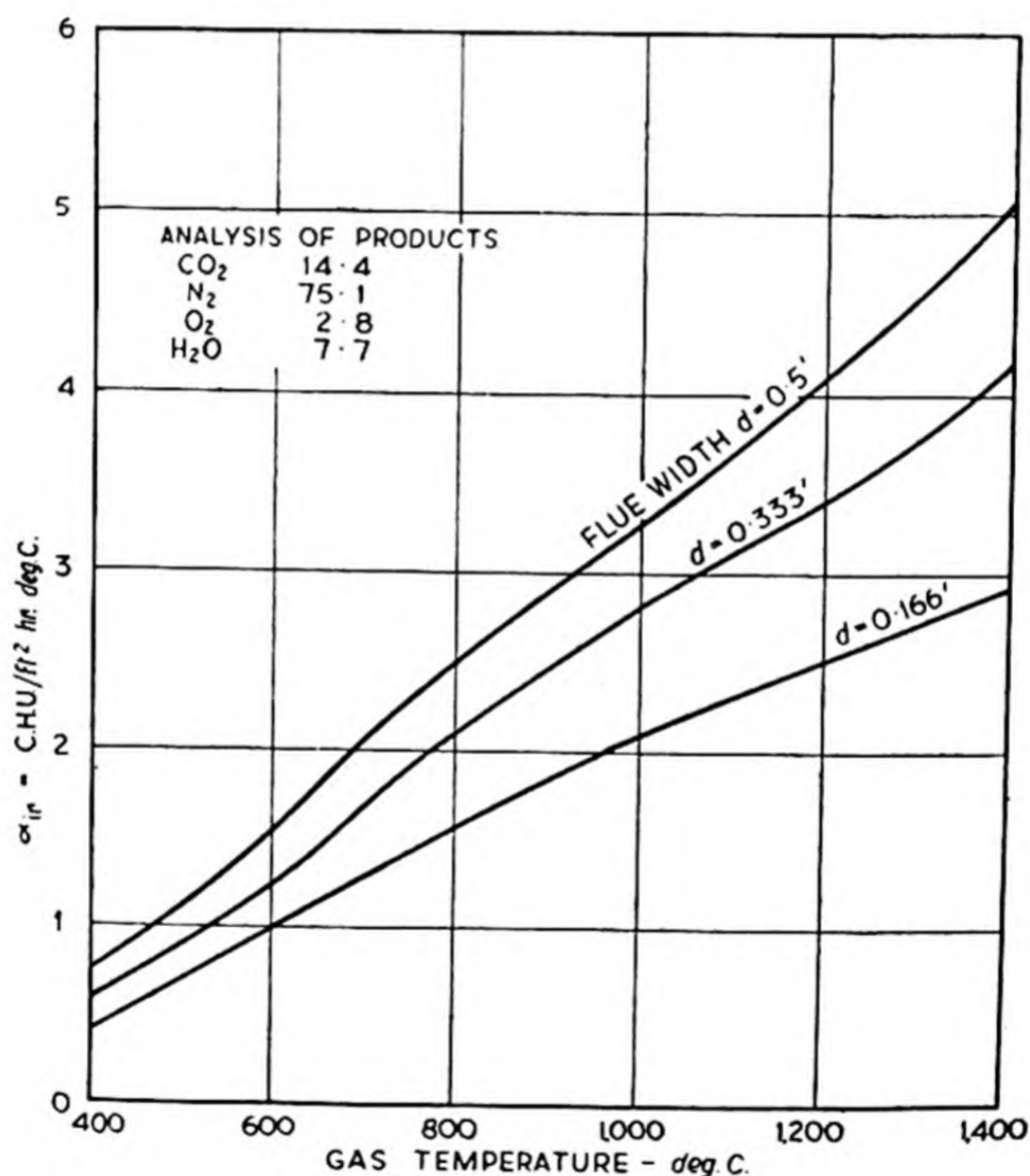


Fig. (4)23. Radiation Heat Transfer Coefficient for Products of Combustion of Producer Gas in Checker Flames.

the products of combustion of producer gas  $\alpha_{1r}$  is given by Fig. (4)23, reproduced by permission from *The Calculation of Open-hearth Furnace Regenerators*, by H. Southern. (*J. Inst. Fuel* 1932, 6, 369).

In the case of  $\alpha_2$  radiation is negligible and the values may be calculated from the formulæ given for  $\alpha_{1c}$ .

Using these formulæ the preheat obtainable in a regenerator and the final gas temperature may be calculated by the formulæ given



for recuperators in 4.3.5 provided  $\alpha$  in the formulæ of 4.5.3 is replaced by  $\phi/t_o$ .

Rummel showed, by means of an experimental regenerator that these formulæ are reliable. From them he concluded that the optimum brick thickness is governed by the practical considerations of survival rather than by the heat transfer requirements, and that there is no great advantage in the use of specially shaped bricks. It should be noted that the formulæ are only applicable to regenerators in which the gas distribution is reasonably uniform—a condition which is frequently not met.

It has been shown from these calculations that a slight overloading of a normal regenerator packing does not greatly raise the gas exit temperature and lower the air exit temperature. The principal inaccuracy in the calculation comes from the values of  $\alpha_1$  and  $\alpha_2$ , which are not known in any given case.

## SUMMARY AND FORMULÆ OF CHAPTER 4.

### HEAT TRANSFER

#### 4.1. Heat Transfer as a Thermodynamic Problem

**4.1.1. The Significance of the Heat Transfer Coefficient.** In order to obtain good fuel economy in a furnace the heating gases must leave the heating chamber at as near the lowest temperature of the charge as possible. To achieve this with a good output the heat transfer coefficient must be as high as possible.

**4.1.2. The Meaning of Temperature.** The temperature conditions at any point in a furnace system can be specified in terms of a gas temperature and a radiant mean temperature both at that point ; these usually differ appreciably.

**4.1.3. Gas Temperature Measurement.** Under these conditions the temperature of a non-luminous gas in a furnace for temperatures up to 1500° C. is best measured with a suction pyrometer, that of a luminous flame with some type of colour pyrometer ; the heat content of gases or flames is, however, more directly observed with a sensible heat meter. The sodium line reversal method is useful for temperature measurement of small non-luminous flames and the Kurlbaum for small luminous flames. A method particularly suitable for measuring the radiant temperature of a flame is the total radiation comparative method.



## 4.2. The Laws of Heat Transfer

**4.2.1. The Mechanisms of Heat Transfer.** Conduction is a handing on of heat energy from molecule to molecule ; convection is the carrying of heat by movement of a fluid ; while radiation is a transmission of vibration energy through empty space.

**4.2.2. Conduction.** For conduction the basic mathematical equation is fully known. It is  $dH = K \cdot \Delta A \cdot dt \frac{\partial T}{\partial n}$ . The difficulties in the way of a full calculation of the heat flow are therefore only mathematical difficulties due to complexity of the shapes.

**4.2.3. Convection.** In the case of convection the mathematical equations cannot even be written down and our knowledge is confined to an extension of experimental results by the theory of similarity. *For natural convection* this gives

$$\frac{HD}{K\theta} = f(\phi \cdot \theta D^3)$$

where  $H$  = heat transfer per unit area and time (C.H.U./ft.<sup>2</sup> hr.) ;

$D$  = characteristic length of the system (ft.) ;

$K$  = thermal conductivity of the gas at the mean gas temperature (C.H.U./ft.<sup>2</sup> ° C. hr.) ;

$\theta$  = temperature difference between gas and solid surfaces (° C.) ;

$\phi$  = a function of the properties of the gas, given in Table 4.1 (col. 4) for  $\phi \cdot \theta D^3 = 10^3 - 10^9$

$$H = 0.47 K \left( \frac{\phi}{D} \right)^{0.25} \theta^{1.25} \quad \text{. . . . . (4/6)(ii)}$$

$$\text{for } \phi \cdot \theta D^3 > 10^9, \quad H = 0.10 \cdot K \phi^{0.33} \theta^{1.33} \quad \text{. . . (iv)}$$

For hot surfaces in atmospheric air these may be simplified to

$$H = 0.35 \cdot \theta^{5/4} \text{ C.H.U./ft.}^2 \text{ hr. (vertical surface) . (4/7)}$$

Horizontal surface facing up—increase  $H$  by 30% ; facing down, decrease by 35%. Horizontal cylinder,  $H = 0.40 \theta^{5/4}$ .

*Forced convection* (where the gas velocity is produced by a cause other than the density changes resulting from the heat flow) :

$$\frac{HD}{K\theta} = f\left(\frac{VcD}{K}\right)$$

where  $V$  = hot gas velocity ;

$c$  = specific heat of gas/unit volume (hot)



for tube bundles

$$H=0.32 C_H \left( \frac{VD}{\nu} \right)^{0.6} \frac{K\theta}{D} \quad \dots \quad (4/9)$$

where  $C_H$  is a numerical shape factor which varies between 0.66 and 1.05 for straight-through tube bundles and between 0.94 and 1.20 for staggered arrangements.

For a gas inside a tube

$$H=0.021 \left( \frac{VD}{\nu} \right)^{0.8} \frac{K\theta}{D} \quad \dots \quad (4/11)$$

**4.2.4. Radiation.** The laws of radiation can not only be expressed mathematically, but they can even be deduced from theoretical considerations. The radiation from a body of emissivity  $E$  at temperature  $T$  ° K. is :

$$R=E\sigma T^4$$

where  $\sigma=1.01 \cdot 10^{-8}$  C.H.U./ft.<sup>2</sup> hr. ° K.<sup>4</sup>

Solids radiate over all wavelengths. Gaseous CO<sub>2</sub> and H<sub>2</sub>O radiate only in bands in the infra-red (non-luminous radiation). To calculate the non-luminous radiation from actual flames, Figs. (4)8, 9 and 10 must be used. Diffusion flames from fuels containing hydrocarbons have a luminosity due to small carbon particles—no formulæ are available, but they are usually much more nearly black ( $E=1$ ).

### 4.3. Heat Transfer in the Furnace Chamber

**4.3.1. Separation of the Heat Transfer Calculation from the Complete Complex Process.** The performance of a heating system depends not alone on the heat transfer but simultaneously on combustion and heat liberation. Hence it is necessary to simplify the calculation in some way. The most practical for furnace calculations is that of Heiligenstadt, where the furnace is divided into steps and an average assumed charge temperature taken in each.

**4.3.2. The Radiation from Actual Furnace Flames.** Since there is considerable doubt in the assumptions necessary to apply the laws of heat transfer to actual furnaces, measurement is always desirable. This may be done by an instrument inserted into the furnace or by an instrument situated in a hole in the wall.

**4.3.3. Convection in the Furnace Chamber.** Empirical values of the convection heat transfer coefficient  $\alpha_c$  in actual furnaces vary over a range from 1 to 10, so that it is preferable to use the formulæ of 4.2.3 which includes the gas velocity as a variable.



#### 4.3.4 THE SCIENCE OF FLAMES AND FURNACES

**4.3.4. The Effect of Muffles on Heat Transfer in the Furnace Chamber.** The use of a muffle to protect the charge has to be paid for by a reduction in the gas-charge heat transfer coefficient and hence by an increased waste-heat loss.

**4.3.5. Parallel and Counterflow Heating.** In a counterflow heater,

$$\text{final temperature of heated gas} = \frac{1}{a - z_o} (T_1(1 - z_o) + T'_2(a - 1))$$

where  $a = \frac{V'c'_p}{\eta Vc_p}$  and

$$z_o = \exp\left(\frac{-\eta\alpha A_o(a - 1)}{V'c'_p}\right);$$

$A$  = total heating area (ft.<sup>2</sup>);

$Vc_p$  = heat capacity of heating gas C.H.U./hr. ° C.;

$V'c'_p$  = heat capacity of heated gas; and

$\eta$  = assumed fraction of heat given up by heating gas which goes to heated gas.

#### 4.4. The Estimation of the Heat Losses through the Furnace Skin in the Steady State

**4.4.1. Methods of Estimating the Heat Losses due to Conduction through the Solid Walls.** The wall losses from an existing furnace are best obtained by measuring the outside wall temperature and using Fig. (4)16 or (4)17. For calculating a prospective furnace they can be estimated from an assumed inside wall temperature together with the resistance of the brickwork and Figs. (4)22 and (4)23.

For the heat losses from mains it is necessary to allow also for the resistance to heat transfer from the gas to the walls.

**4.4.2. Other Causes of Heat Loss.** Heat losses through open doors are calculated from the black-body radiation multiplied by a factor which varies between 0.1 for small holes in thick walls to 1.0 for the converse. Heat losses to water-cooled pipes can be calculated from radiation and conduction formulæ.

#### 4.5. Heat Flow in the Unsteady State

**4.5.1. General Methods.** Unsteady-state heat problems (heating up of charge in batch furnaces and heat flow into walls) can be solved by detailed calculation, by approximate methods of calculation or by the use of similarity from models.



**4.5.2. The Heating up of the Charge in a Batch-type Furnace.** The temperature  $T_c$  of a thin body heated in a batch furnace which is at constant temperature  $T_f$  after time  $t$  is given by

$$T_c = T_f - (T_f - T_o) \exp \left( \frac{-\alpha t}{gc} \right) \quad . \quad . \quad . \quad (4/31)$$

For thick bodies the time is longer, since the effective value of  $\alpha$  is reduced by the thermal resistance of the charge.

### 4.5.3. Furnace Walls

Total heat lost in a shut-off period =  $Q_m[1 - \exp(-t_o/t_m)]$  (4/35)

where  $Q_m$  = total heat stored in brickwork ; and

$t_m = Q_m/H_o$ , where  $H_o$  = rate of heat loss in the steady state just before the shut-down.

**4.5.4. Regenerators.** Regenerators may be calculated in the same way as recuperators (Chapter 4.3.5) provided that the coefficient  $\alpha$  is replaced by a coefficient  $\phi$  given by :

$$\frac{1}{\phi} = \frac{1}{\alpha_1 t_1} + \frac{1}{\alpha_w t_o} + \frac{1}{\alpha_2 t_2}$$

where  $\alpha_w$  is a factor for the heat transfer into the brickwork (=0.0051 C.H.U./ft.<sup>2</sup> hr. ° C. in most cases) ;

$t_o = t_1 + t_2$  and  $t_1$  = time on waste gases (hrs.),  $t_2$  = time on air (hrs.) ; and

$\alpha_2$  is the brick/air convection coefficient  $\left( \frac{1.26 V_o^{1/2}}{d^{1/3}} \right)$  where  $V_o$  = N.T.P. velocity of gas in checker ft./sec.  $d$  = side of square flue in ft.) and  $\alpha_1$  is given by this value for convection plus that for radiation (see Fig. (4)23).



## **5. THE AERODYNAMICS OF HOT SYSTEMS**

### **5.1. The Factors Governing the Total Quantity of Gas flowing in a Circuit. Fundamental Principles**

**5.1.1. The Significance of High-temperature Aerodynamics.** In the preceding two chapters of this monograph it has been shown that control of combustion and of heat transfer in a furnace are essential to the achievement of the aims of high output and efficiency. Combustion and heat transfer are themselves strongly affected by the flow of the gases in a heating system and hence this flow must also be carefully studied. In a cold system the flow pattern is governed by two internal forces, viscosity or fluid friction and inertia, while the total quantity flowing is determined by the total fluid friction and the applied aeromotive force\* derived from a source of power such as fan, pump or injector. In a hot system, on the other hand, an extra factor enters in deciding both the flow pattern and the total quantity flowing. This extra factor is the change of density of gases with change in temperature which results in the appearance of buoyancy forces. Since the buoyancy forces completely alter both the flow pattern and the laws governing the total quantity flowing, the term "high-temperature aerodynamics" is introduced to describe the flow of gases influenced by these forces.

An example will serve to bring out the importance of density effects in a heating system. When gases flow through a horizontal channel all the walls of which are at the same temperature a certain amount of mixing occurs between the gases from the top and bottom of the stream as a result of diffusion and turbulent motion. If, however, relatively cold material to be heated is placed along the floor of the channel, the gases which first come into contact with the material are cooled. They are then heavier than those above them and hence the action of gravity reduces the amount of mixing between the top and bottom of the stream. This reduces the

\* In describing the flow processes in a furnace system implicit use will be made of the electrical analogy which is dealt with more fully in 5.1.4 below. The term "aeromotive force" is a convenient one to be equivalent to the electromotive force in a circuit as that which makes the current flow. It includes fans, buoyancy in a vertical column of hot gases, injectors and volume changes resulting from chemical reaction.



amount of heat transferred to the cold material considerably compared to the amount which would be transferred if the flow were governed by viscosity and inertia alone.

Conversely, if the cold charge could be arranged along the roof, the action of gravity would increase the mixing and hence the rate of heat transfer.

It will be shown in this chapter that while the flow of cold systems has been very fully studied, particularly in connection with aircraft and with flow in tubes, the laws of "high-temperature aerodynamics" are not nearly so well explored.

**5.1.2. The Two Basic Problems of High-temperature Aerodynamics.** As has already been suggested, there are two distinguishable problems of hot aerodynamics each of which is of great importance to the achievement of good furnace performance. In the first place it is necessary to know how to control, calculate and measure the total quantity of air and gas which will flow through the system under given conditions.\* This quantity determines the amount of fuel which will be burnt and the quality of the combustion. Secondly, it is necessary to have the same kinds of knowledge of the inner structure of the gas flow, that is to say, knowledge of the laws governing the velocity distribution in wide channels, the formation of stagnant pockets, of large eddies, the persistence of jets, mixing of streams, and the subdivision of the flow between channels connected in parallel.† This second problem is of great importance where it is necessary (1) to obtain by convection a high rate of heat transfer from the gases to the material being heated, since this can only be achieved by attaining a maximum gas velocity in the neighbourhood of the charged material ; (2) to obtain a good utilisation of the whole of a distributed heat transfer surface such as the packing of a regenerator ; and (3) to control the volume required for combustion in gas-air diffusion flames, oil flames and pulverised coal flames which, as shown in Sections 3.3, 3.4 and 3.5, is predominantly a matter of controlling the mixing rate. The most important difference between these two problems is that the first is susceptible to approximate calculation whereas the equations governing the second cannot be solved and our knowledge of it

\* It is convenient to consider all three requirements together, although the designer usually wants to know *how to calculate* the design which will give the conditions he wants, whereas the furnace user needs to know *how to control* the furnace to give these conditions and *how to measure* whether he is getting them.

† See footnote to p. 263.



### 5.1.2 THE SCIENCE OF FLAMES AND FURNACES

comes mainly from applying model results by means of similarity theory. The two problems are thus closely analogous in this respect to heat conduction and convection respectively. The first five sections of this chapter, 5.1, 5.2, 5.3, 5.4 and 5.5, refer to the first of these problems, the remaining two sections, 5.6 and 5.7, to the second.

**5.1.3. The Laws Governing the Total Quantity Flowing in a Channel.** For the moment attention is confined to the flow through a single-channel system. This channel will have various bends, changes of section and of temperature, and will sometimes run horizontally, sometimes vertically or obliquely. Subdivisions of the flow will be considered later.

Before going further it is necessary to clarify the idea of pressure. The true static pressure at a point in a fluid is defined as the mean pressure exerted over the surface of a hypothetical small body moving with the fluid. Except in the case of very viscous liquids, with which we are not concerned, this pressure would be uniform over such a surface and the word "mean" may be omitted from the definition.

This pressure  $p'$  corresponds to the barometric pressure in still air and varies with height above sea-level. Pressures in flow measurement are not measured barometrically but by manometers, which measure difference from atmospheric pressure at the same height ( $h$ ). We therefore introduce the expression  $p = p' + g\rho_0 h$ , when  $\rho_0$  is density of air at room temperature.  $p$  is the same at all points in still air at room temperature and it is the change in  $p$  along the flow path which we shall calculate.

As discussed in Section 5.1.5, the flow in furnace ducts is generally turbulent, i.e. the velocity at a point varies rapidly about a mean value. The fluctuations are important to the theoretical aspect of flow, but practically it is only the mean velocities which concern us. Lines drawn so that at every point each line lies in the direction of the mean velocity are called stream lines and have by definition the property that there is no mean flow across a stream line. Suppose a small area is chosen perpendicular to the flow and the stream lines which pass through its perimeter are drawn. These enclose a tube known as a stream tube (Fig. (5)1) and through it flows the same amount of fluid as crosses the original area. ✓

As an element of fluid moves down a stream tube it is acted on by various forces which may be resolved parallel and perpendicular to its motion. The perpendicular components keep it to its path



but do no work on it nor alter its speed. We shall now consider the parallel components. The forces are :

- (1) Its weight,  $-\rho g dA ds \frac{dh}{ds}$  ;
- (2) gradient of true static pressure,  $-dA ds \frac{dp'}{ds}$  ; and
- (3) shear forces over the sides of the tube,  $-B dA ds$ .

where  $B$  is the shear force per unit volume ;

$ds$  is the length of the element ;

$dA$  is the area of the tube ; and

$g$  is the gravitational constant.

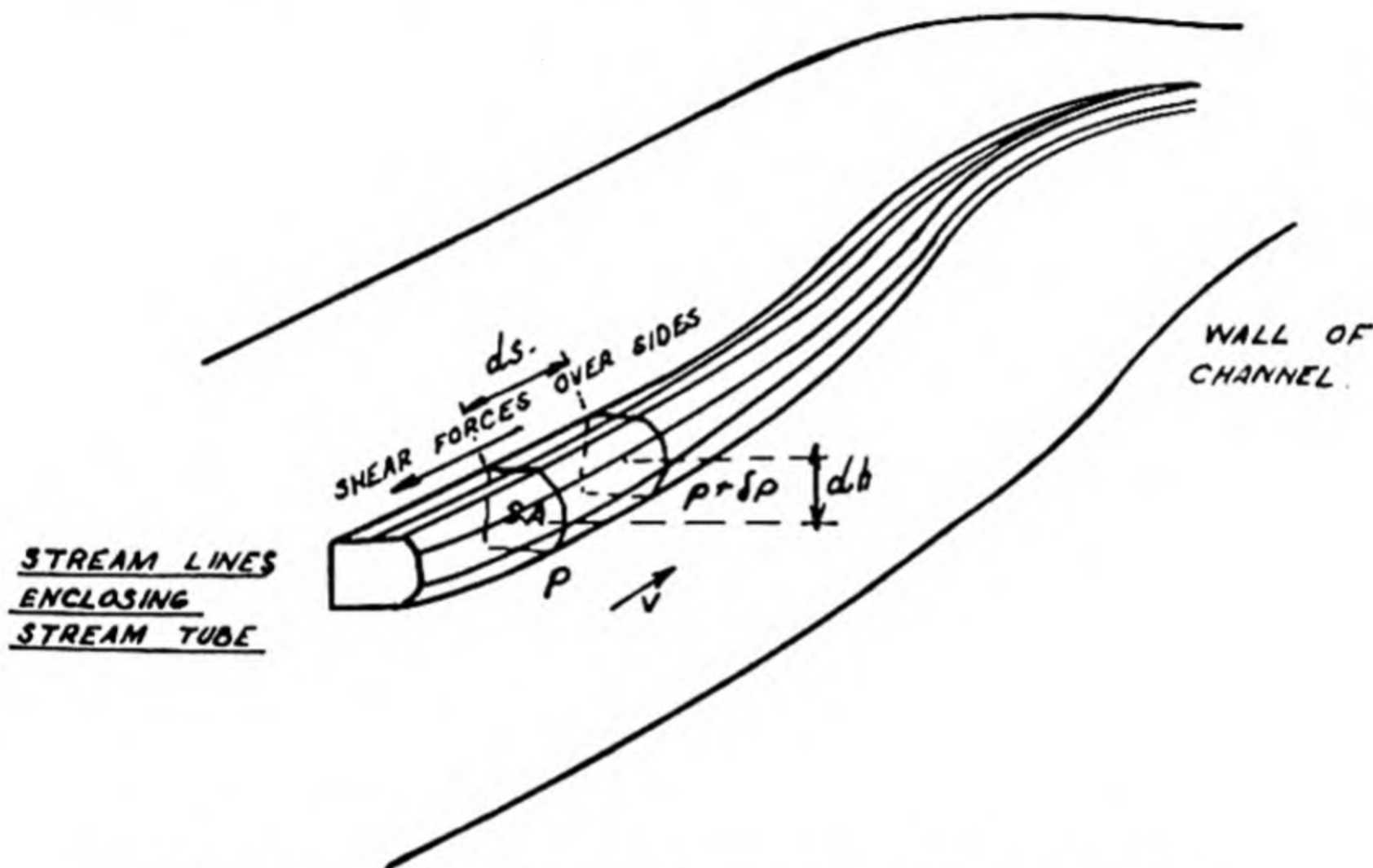


Fig. (5)1. Element of Fluid Moving Along Stream Tube.

Equating these to the mass  $\times$  acceleration

$$\rho dA ds \frac{dv}{dt} = \rho dA ds v \frac{dv}{ds} = -\rho g dA ds \frac{dh}{ds} - dA ds \frac{dp'}{ds} - B dA ds$$

$$\text{i.e. } \rho v \frac{dv}{ds} + \rho g \frac{dh}{ds} + \frac{dp'}{ds} + B = 0 \quad (5/1)$$

putting  $p' = p - g\rho_0 h$ .

$$\rho v \frac{dv}{ds} + \rho g \frac{dh}{ds} + \frac{dp}{ds} - \rho_0 g \frac{dh}{ds} + B = 0$$

Integrating along the stream line

$$\Delta p \equiv p_2 - p_1 = -g \int (\rho - \rho_0) dh - \int \rho v dv - \int B ds \quad (5/2)$$

We shall call the three terms on the right-hand side  $\Delta p_B$ ,  $\Delta p_K$ ,  $\Delta p_R$



### 5.1.3 THE SCIENCE OF FLAMES AND FURNACES

respectively, for they represent buoyancy, conversion of kinetic energy and resistance loss.

$$\text{Thus} \quad \Delta p = \Delta p_B + \Delta p_K + \Delta p_R \quad . \quad . \quad . \quad (5/3)$$

(a) *Buoyancy.* The horizontal parts of the path make no contribution to  $\Delta p_B$ . For a part where the density is constant  $\Delta p_B = -g(\rho - \rho_0)\Delta h$ . If, as in a checker, the density varies over the height an estimated mean value of  $\rho$  should be used. The formula may be modified for use in practical units.

$$\begin{aligned} \Delta p_B &= -g\rho_0(\rho/\rho_0 - 1)\Delta h \quad . \quad . \quad . \quad (5/4) \\ &= -g\rho_0\left(\frac{288\delta}{273+T} - 1\right)\Delta h \end{aligned}$$

Taking atmospheric temperature as 15° C. (59° F.) :

$$\Delta p_B = 0.0147\left(1 - \frac{288\delta}{273+T}\right)\Delta h \quad . \quad . \quad . \quad (5/5)$$

where  $p_B$  is in inches w.g. ;

$h$  in ft. ;

$\delta$  density of gas relative to air at same temperature ; and  
 $T$  is temperature of gas in ° C.

(b) *Kinetic energy conversion.* We have from (5/2)

$$\begin{aligned} \Delta p_K &= -\int \rho v dv \quad \therefore dp_K = -\rho v dv \\ \int \frac{dp_K}{\rho} &= -\int v dv = -\Delta v^2/2 \quad . \quad . \quad . \quad (5/6) \end{aligned}$$

This is the general form of Bernoulli's theorem for compressible fluids.

The integral on the left hand can only be evaluated exactly if the variation of  $1/\rho$  with  $p_K$  is known. But if the path is divided into sections so small that changes in  $1/\rho$  are approximately proportional to changes in  $p_K$  we have, by substituting the mean value for  $1/\rho$  :

$$\begin{aligned} \Delta v^2/2 &= -\frac{1}{2}(1/\rho_1 + 1/\rho_2)\Delta p_K \\ \therefore \Delta p_K &= -\frac{\rho_1\rho_2}{\rho_1 + \rho_2}\Delta v^2 \quad . \quad . \quad . \quad (5/7) \end{aligned}$$

When  $\rho$  is constant this reduces to

$$\Delta p_K = -\rho\Delta v^2/2 \quad . \quad . \quad . \quad (5/8)$$

This is the simplified form of Bernoulli's equation for a fluid whose density does not change appreciably in the length of channel concerned. The expression on the right hand is called the change



in the dynamic head.\* The term  $\Delta p_K$  evaluated by this equation is important where the flow narrows down to a small cross-section and the velocity is high, as for example in the gas port of a producer-gas-fired furnace, in the discharge at the top of a chimney or in the throat of a venturi.

$p + \rho v^2/2$  is called the total head in the system and if the pressure

TABLE 5.1  
CALCULATED DYNAMIC HEADS  $\frac{1}{2}\rho v^2$ , FOR AIR  
(in ten-thousandths inch water-gauge)

Pressure corresponding to	Temperature ° C.						
	0	30	100	200	300	400	600
Actual velocity 1 ft./sec. N.T.P. velocity 1 ft./sec.	2.41	2.17	1.76	1.39	1.15	0.98	0.75
	2.41	2.67	3.29	4.17	5.05	5.94	7.70
	800	1000	1200	1400	1600	2000	
Actual velocity 1 ft./sec. N.T.P. velocity 1 ft./sec.	0.61	0.52	0.45	0.39	0.35	0.29	
	9.47	11.20	13.00	14.80	16.50	20.10	

This table refers to actual or N.T.P. velocities of 1 ft./sec. for air. For gas of  $s$  times the density of air at the same temperature moving with velocity  $v$  ft./sec. the dynamic head is  $sv^2$  times that tabulated.

balance is constructed for the total head instead of the static pressure  $p$ , then the changes in the dynamic head are automatically allowed for. If a straight tube connected to a manometer is arranged with its open end facing directly upstream the pressure measured is the total head,  $p + \rho v^2/2$ .† If the static head can also be measured, the difference is the dynamic head from which the velocity can be calculated. This is the principle of the well-known

\* The dynamic head is only numerically equal to  $\rho v^2/2$  when expressed in consistent units. Table 5.1 gives the dynamic head in common units.

† This is shown by applying equation (5/3) to a short length of stream line upstream from the tip. Over this short distance change in density and friction losses are negligible and  $\Delta p = \Delta p_K = -\Delta \frac{\rho v^2}{2}$ . Since  $v=0$  at the tip of the tube, if  $p_H$  is the pressure in the tube and  $p$  the static pressure upstream,  $p_H - p = -\left(0 - \frac{\rho v^2}{2}\right)$  from (5/8) and so  $p_H = p + \frac{\rho v^2}{2}$ .



### 5.1.3 THE SCIENCE OF FLAMES AND FURNACES

pitot static tube. Now there are certain cases in furnace systems where  $\rho$  changes significantly in the section concerned, e.g. in the heating in a checker or in actual combustion. In general these cases cannot be calculated from the fundamental laws of dynamics unless the distribution of  $\rho$  is known, and even then experimental evaluation is usually necessary. There are two cases, however, in which (5/6) can usefully be applied with varying  $\rho$ , in both of which the channel has a constant cross-section.

When  $\rho$  changes during flow along a channel of constant cross-section so that the product  $\rho v$  which is the mass flow rate divided by the cross-section area is constant we have from (5/7)

$$\Delta p_K = -\frac{\rho_1^2 v_1 / v_2}{\rho_1 + v_1 / v_2 \rho_1} (v_2^2 - v_1^2) \text{ since } \rho_1 v_1 = \rho_2 v_2$$

$$= -(\rho_2 v_2^2 - \rho_1 v_1^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5/9)$$

In this special case the answer is exact and could have been obtained by integrating  $dp = -(\rho v)dv$ , where  $\rho v$  is constant.

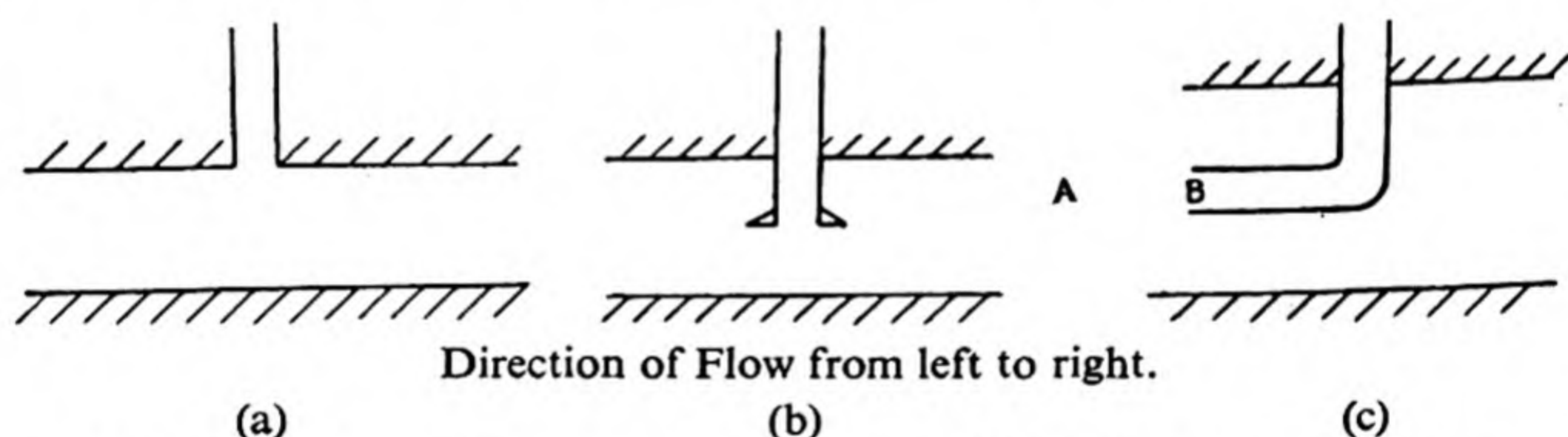


Fig. (5)2. Types of Pressure Tapping. (a), (b) Static Pressure Tapping, (c) Total Pressure.

Equation (5/9) is sometimes known as the momentum equation, as it can be obtained by consideration of the conservation of momentum. It is important to realise, however, that the law of conservation of momentum cannot be simply applied to a channel of varying cross-sectional area, as the normal pressure on the walls contributes to the momentum in the latter case.

The results obtained so far refer to values along a stream tube. But the whole channel consists of a number of such tubes lying side by side completely filling it without overlapping, and if the equations for each tube are imagined added together and divided by the number of tubes the result is a similar equation in which each value is the mean across the section. It is in this form that we shall use them, for the pressure across a section is uniform except for buoyancy corrections if there is no curvature of the stream lines.



In such a straight section the static pressure may be measured by inserting a tube flush with the wall which measures the static pressure at the wall and correcting for buoyancy pressure to the centre of the channel. The formula for buoyancy pressure and kinetic energy conversion pressure which we have shown to hold along a stream line can thus be used along a channel.

The other case which can sometimes be usefully treated by applying equation (5/9) is the mixing of two streams which flow parallel before and after mixing in a channel with parallel walls, as for example in the throat of an injector or where a jet entrains surrounding gas in a large chamber. In this case it is always necessary to make allowance for frictional losses at the same time, and this allowance is usually empirical and hence takes account of the lack of exact fulfilment of the requirements for applicability of the law.

In applying the equation of flow in a channel the following expressions of the gas laws and the law of conservation of matter are convenient.

Let the cross-sectional area of the channel be	$A$ ft. <sup>2</sup> at any point.
the gas flow quantity be	$Q$ ft. <sup>3</sup> /sec. at any point.
the gas flow quantity at N.T.P. be	$Q_0$ ft. <sup>3</sup> /sec.
the gas flow mass flow be	$M$ lb./sec.
the gas flow velocity be	$V$ ft./sec. at any point.
the gas density be	$\rho$ lb./ft. <sup>3</sup> at any point.
the absolute gas temperature be	$T$ ° K.

$$\left. \begin{aligned} \text{Then } \rho &= \frac{M}{Q} & \rho_0 &= \frac{M}{Q_0} \\ V &= \frac{Q}{A} = \frac{M}{\rho A} = \frac{\rho_0 Q_0}{\rho A} & \frac{\rho}{\rho_0} &= \frac{T_0}{T} \end{aligned} \right\} \dots \dots \dots (5/10)$$

$$\frac{Q_1}{Q_2} = \frac{\rho_2}{\rho_1} = \frac{T_1}{T_2}$$

neglecting the effect of pressure on volume, as may be done in furnaces in nearly all cases.

Thus the flow conditions can be fully calculated if  $A$  and  $T$  are given at all points, the formula for the gas density in terms of temperature is known, and any one of  $Q$ ,  $M$ ,  $Q_0$  or  $V$  is known at any one point in the flow system.

(c) *Friction loss.* The evaluation of  $\Delta p_R$  cannot be done from



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theory. But it can be shown by dimensional analysis that for any given geometrical shape

$$-\Delta p_R = \frac{\rho v^2}{2} f(R_e) \quad . \quad . \quad . \quad . \quad . \quad (5/11)$$

where  $R_e$  is the Reynolds number.

Experiment shows that if  $R_e$  is large then  $f(R_e) = F$ , where  $F$  is a constant known as the friction factor. The friction factors of many simple shapes (changes of section, bends, etc.) may be found in the literature, and failing this, if experience does not provide a sufficiently close estimate, they may be determined by a small-scale model at room temperature. They are discussed in detail in section 5.3.

Friction factors are conventionally expressed as the ratio of the friction loss  $-\Delta p_R$  to the dynamic head  $\frac{\rho_0 v_0^2}{2}$ , where  $\rho_0$  and  $v_0$  are the density and mean velocity at the narrowest point of the part of the channel to which they refer. For example, in applying the friction factor to a sharp expansion the value of  $v$  before the expansion is to be used.

Returning to equation (5/3)

$$\Delta p = \Delta p_B + \Delta p_K + \Delta p_R$$

We have seen that  $\Delta p_B$  can be evaluated if the path is broken up into sections in which  $\rho$  is constant or varies nearly proportionally to height.  $\Delta p_K$  can be evaluated for sections in which  $1/\rho$  is constant or varies nearly proportionally with  $p$ , and  $\Delta p_R$  for sections in which  $\rho$  does not vary much. Writing  $\Sigma \Delta p_B$ ,  $\Sigma \Delta p_K$ , or  $\Sigma \Delta p_R$  for the sum of all such sections we have

$$\Delta p = -\Sigma g \rho_0 (\rho/\rho_0 - 1) \Delta h - \Sigma \frac{\rho_1 \rho_2}{\rho_1 + \rho_2} \Delta v^2 - \Sigma F \frac{\rho v^2}{2} \quad (5/12)$$

where the subscripts 1, 2 refer to the beginning and end of the sections and in the last term  $\rho v^2/2$  is expressed for the narrowest part of the section.

(d) *Junction between two streams.* At a junction between two streams the quantity flowing in the combined stream is obviously equal to the sum of those in the separate streams so that the flow rate and hence the dynamic head in the combined stream is known in terms of the flow rates in the separate streams. There is also continuity of pressure, so that the pressures of the three streams differ by the pressure changes at the junction which are proportional



to the dynamic heads of the two streams. The factors of proportionality depend on the shape of the junction and must be determined by experiment, for example with models, or obtained from tables (see Sections 5.6 and 5.7).

If the resistance of each path is large compared with the dynamic heads at the junction, the pressure changes at the junction may be neglected. In either case when the equations of the type of 5/12 are set up for the pressure changes along each path through the junction and along the combined path, then if the dynamic heads are expressed in terms of  $m_1$ ,  $m_2$ ,  $m_3$  (the mass flow rates in the branches and common channel) two equations are obtained of the form.

$$\left. \begin{aligned} Am_1^2 + Bm_2^2 + Cm_3^2 &= D \\ A'm_1^2 + B'm_2^2 + C'm_3^2 &= D' \end{aligned} \right\} \quad \cdot \quad \cdot \quad \cdot \quad (5/13)$$

*Note :* The  $m_2^2$  terms in the equation referring to channel 1 may arise from the junction pressure terms.

where  $A B C D$ ,  $A' B' C' D'$  are some constants which are obtained by the methods described above. These equations may be handled in the following way.

Solving for  $m_1^2$  and  $m_2^2$  in terms of  $m_3^2$  two equations of the following form are obtained :

$$\left. \begin{aligned} m_1^2 &= Em_3^2 + F \\ m_2^2 &= Gm_3^2 + H \end{aligned} \right\} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (5/14)$$

where  $E F G H$  are constants. But we have also  $m_1 + m_2 = m_3$ .

These three equations may be solved most easily by plotting graphically  $m_1 + m_2$  calculated from (5/14) against  $m_3$ . The point where this curve meets the straight line of  $m_1 + m_2$  calculated from  $m_1 + m_2 = m_3$  (at  $45^\circ$  through the origin) gives the value of  $m_3$ . Hence equations (5/14) give  $m_1$  and  $m_2$ . By this means much algebraic manipulation is avoided.

**5.1.4. The Electrical Analogy.** In introducing the term "aero-motive force" use has already been made of an analogy which is of value in considering the total flow of air and gas in a system. This is the analogy with the flow of electric current in a circuit.\* The analogy is of value because it supplies concepts and a diagrammatic

\* It has been pointed out that in the early days of electrical engineering the flow of electricity was frequently described by analogy with the flow of liquids or gases in tubes, whereas now the concepts of electricity are so familiar that the wheel has come full circle and it is found to be useful to borrow them to describe the flow of gases in systems.



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representation which are of great convenience in thinking, even though it is incomplete. The analogy is given in Table 5.2.

TABLE 5.2  
THE ELECTRICAL ANALOGY

	Aerodynamic term	Electrical analogue
1	The atmosphere	Earth.
2	Section of channel	Conducting wire.
3	Total pressure at any point relative to atmospheric pressure at the same height	Potential relative to earth.
4	Buoyancy } Producing Injectors } "aeromotive Fans } force"	Electric batteries or dynamos producing E.M.F.
5	Resistance to flow due to friction along walls and at bends	Electrical resistance = $R$ .
6	N.T.P. quantity of gas flowing in one section of channel $Q_0$	Electric current, $C$ .
7	Law of resistance, $\Delta p_R = KQ_0^2$	Ohm's law. Potential drop across resistance = $CR$ .

Two examples will serve to show how the analogy enables the diagrams of the electric circuit to be used to represent air-flow circuits.

*Example 1.* Fig. (5)3a is a vertical tube partially closed at the top and bottom, air which enters at the bottom being heated to a temperature appreciably above that of the surroundings. Since the

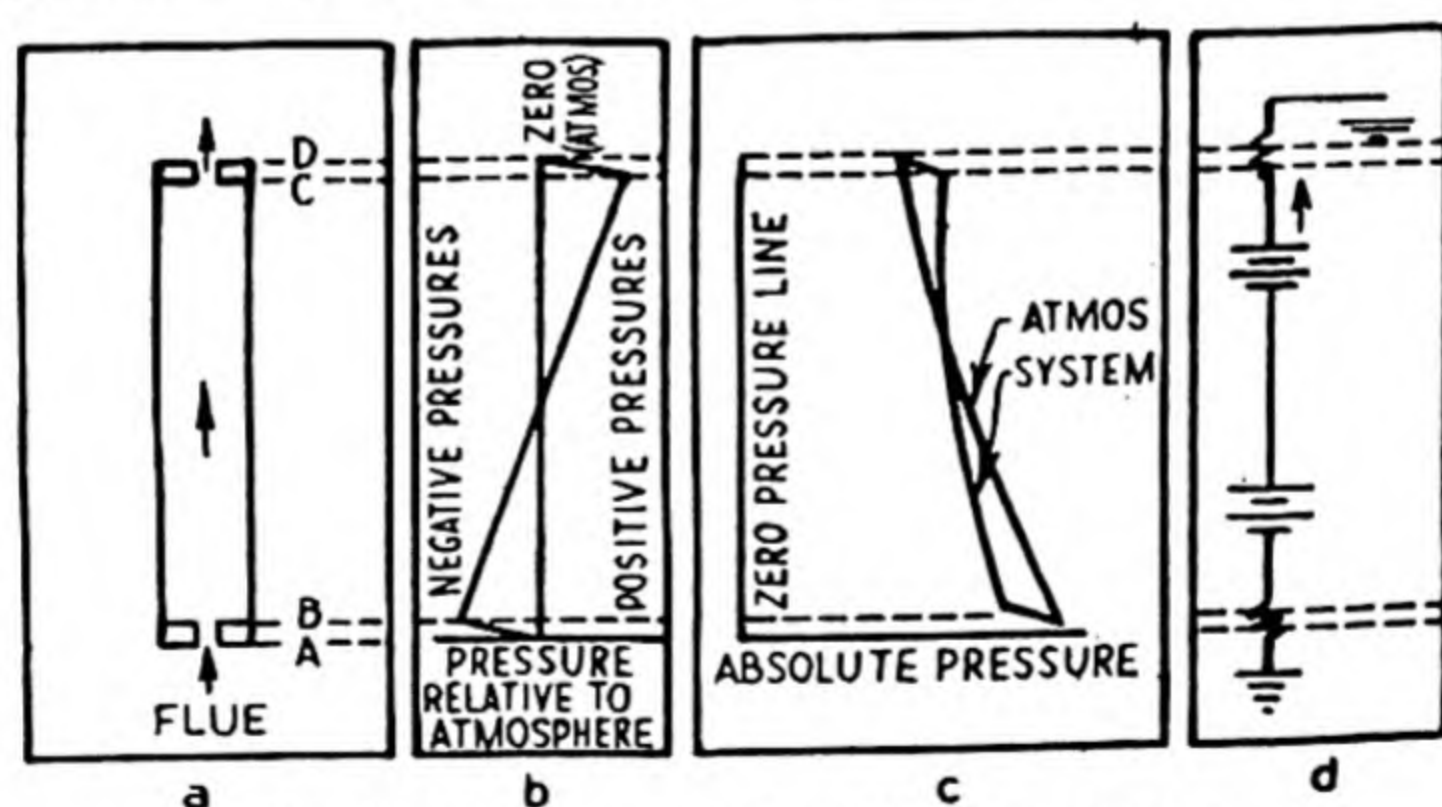


Fig. (5)3. Buoyancy in a Vertical Tube.

column of hot air inside is lighter than that of cold air outside, the absolute pressure inside falls off less than that outside (Fig. (5)3b) and a buoyancy pressure  $\Delta p_B$  relative to atmosphere (see Fig. (5)3c) is therefore set up, as discussed in the last section.  $\Delta p_B$  is positive since it tends to make the pressure higher at a point subsequent in the flow. The buoyancy pressure will set up a flow, and when this has become steady the total pressure loss due to resistance,  $\Delta p_R$ ,



which increases with the air velocity, will be equal to the buoyancy pressure. The equivalent electric circuit is illustrated in Fig. (5)3d. An electric battery with the positive pole at the top is connected by a resistance at each pole to the earth and causes a current to flow through the resistances.

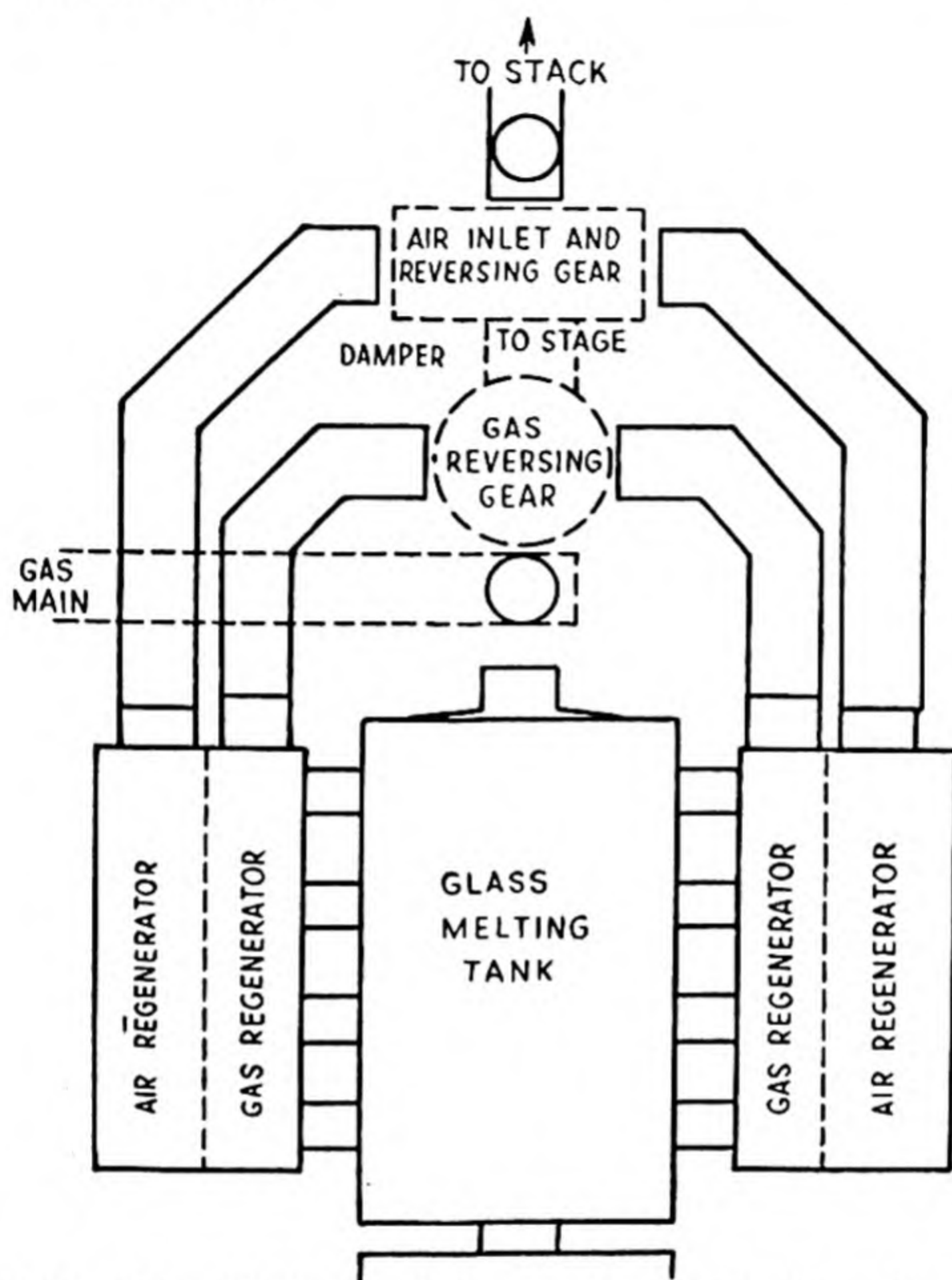


Fig. (5)4a. Glass Melting Tank showing Air and Gas Flow Circuits.

It is of interest that in this case a manometer connected across  $BC$  will indicate the total buoyancy pressure  $\Delta p_B$  because there is no appreciable resistance in this part of a circuit, just as a voltmeter connected across a battery of zero internal resistance will read the E.M.F. of the battery even when a current is flowing.

*Example 2.* Fig. (5)4a is a glass tank system consisting of four regenerators connected by ports to the melting tank and by mains to reversing valves and thence to the stack and gas-producer. Cold air and relatively cool producer gas enter at the base of two regenerators on one side and pass upwards while being heated. They



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burn in the melting tank and the hot waste gases pass down through the opposite regenerators to the stack. An investigation<sup>5.1</sup> showed that the principal resistances to flow occurred in the reversing valves and in the ports, whereas the sources of aeromotive force were the buoyancy in the regenerators and in the stack.

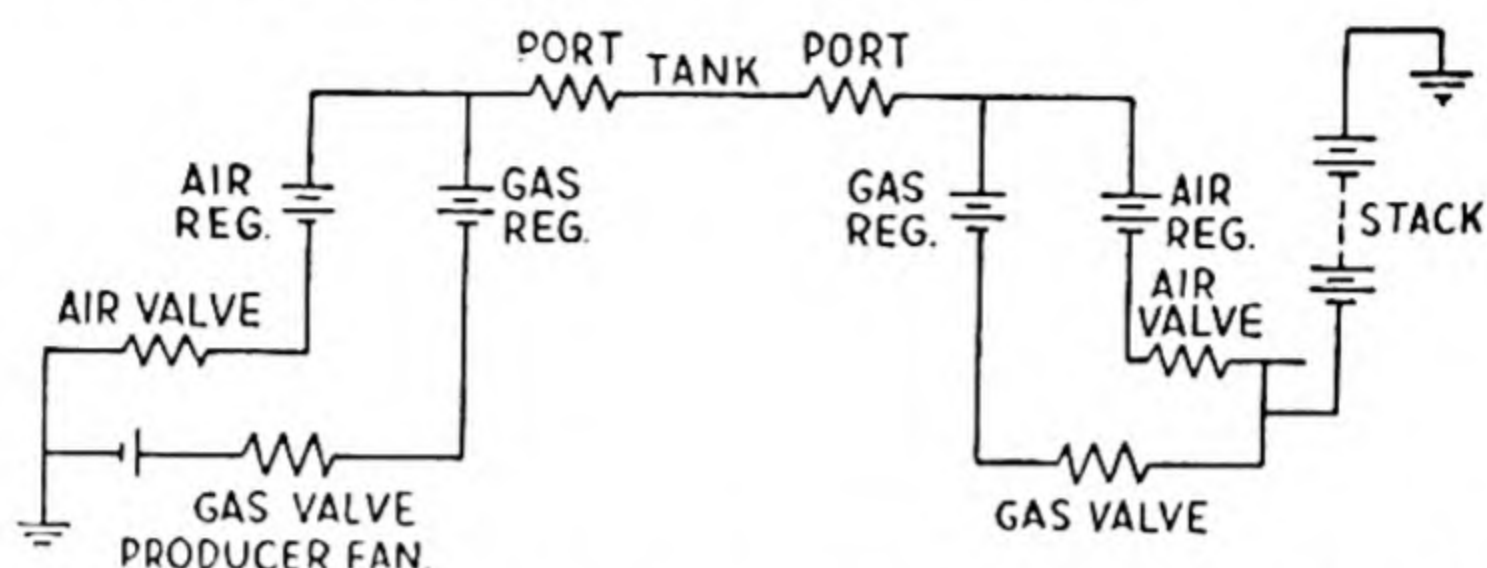


Fig. (5)4b. Electrical Analogue for Gas and Air Flow in Glass Melting Tank System of Fig. (5)4a.

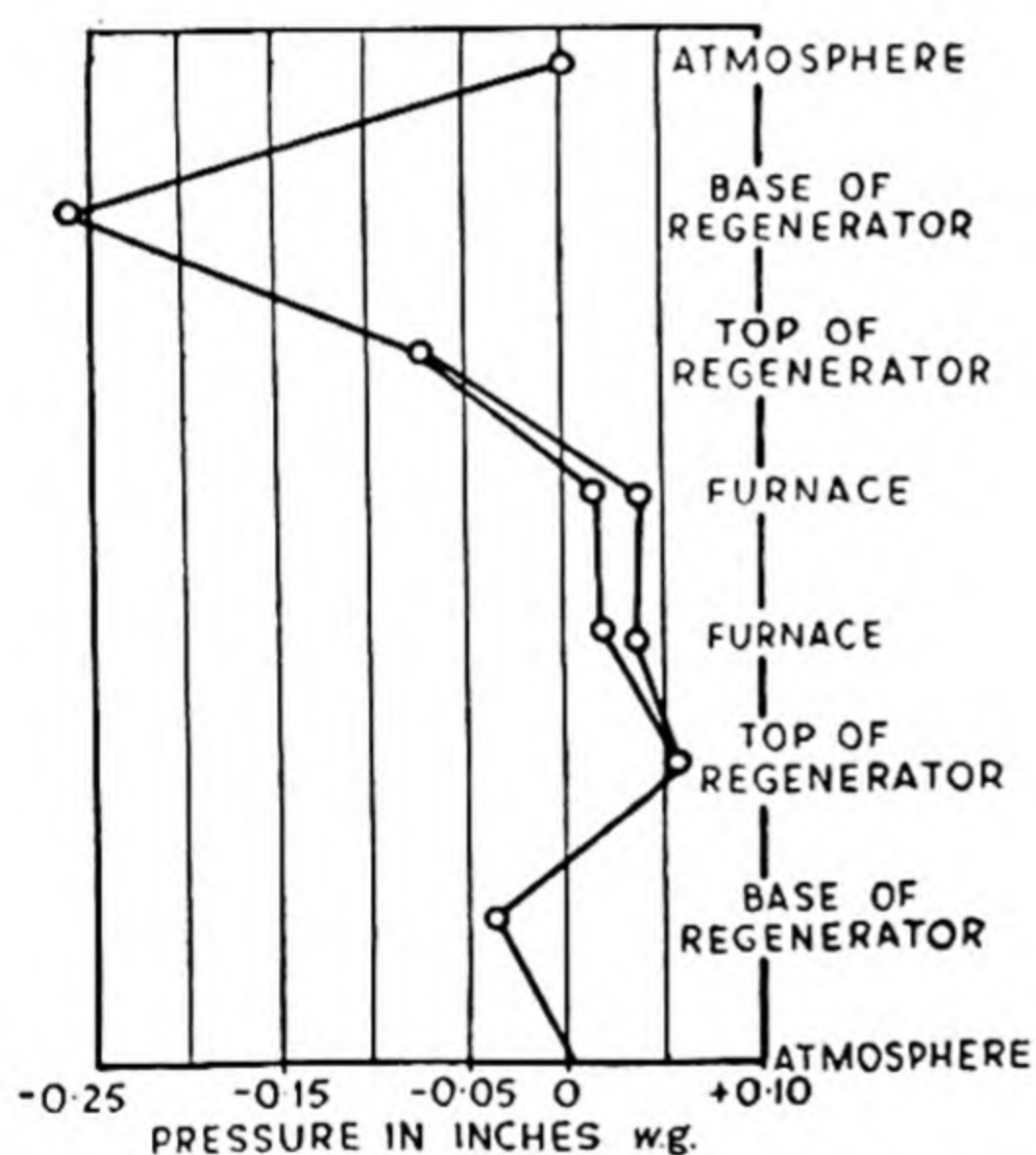


Fig. (5)4c. Method of representing Static Pressures through Furnace System Applied to Air Flow System of Glass Tank.

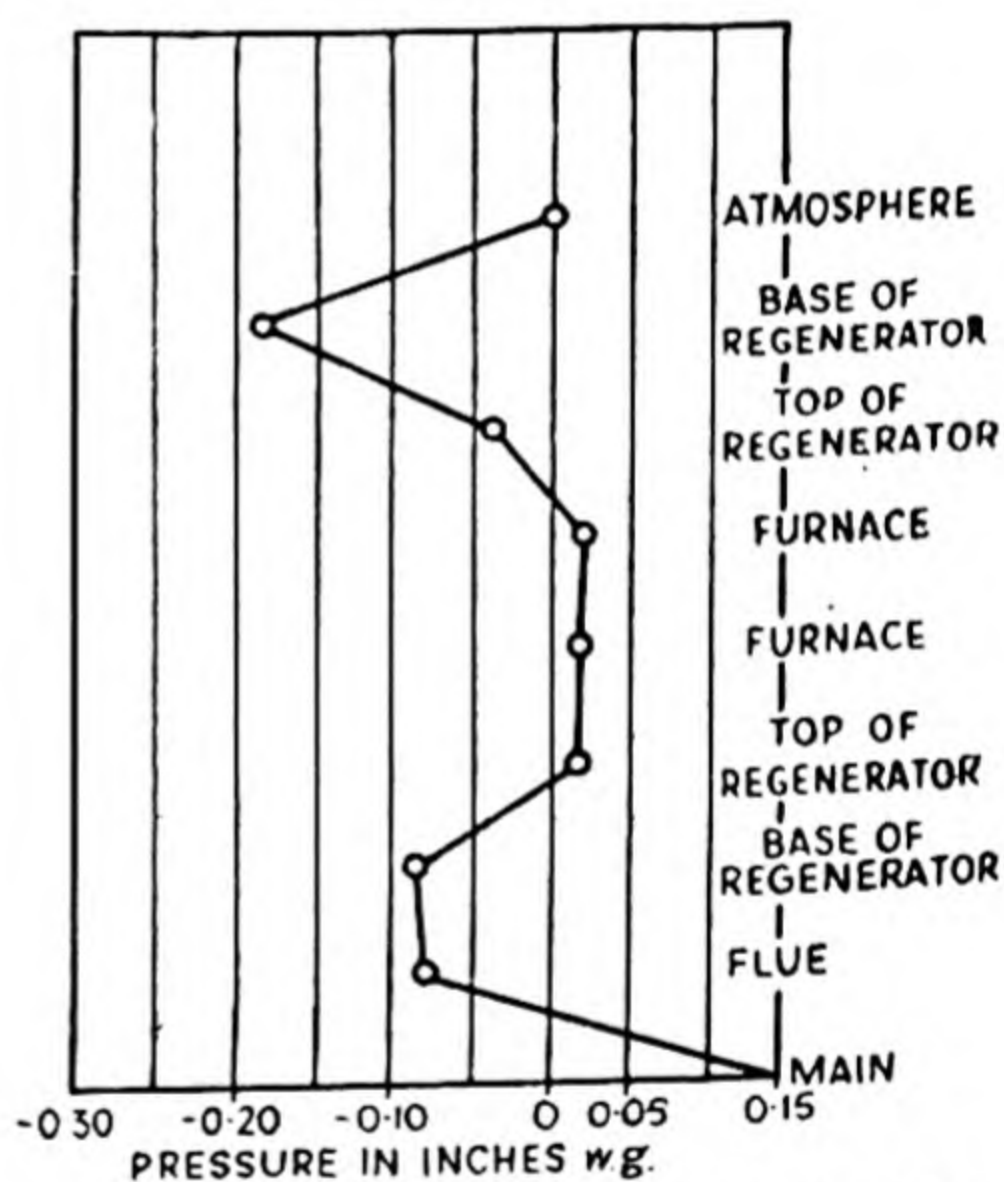


Fig. (5)4d. Method of representing Static Pressures through Furnace System applied to Gas Flow System of Glass Tank.

The equivalent electrical circuit is shown in Fig. (5)4b. The diagram illustrates a point which is of great importance in furnaces of this type. This is that the existence of the "battery," or buoyancy in the ingoing regenerators, enables the flow to be obtained through the resistance of the ingoing valves and ports while the pressure in the furnace is maintained very nearly at atmospheric. There is of

5.1 GOODING, E. J., and THRING, M. W.: "The Flow of Gases in Natural Draught Furnaces," *Trans. Soc. Glass Tech.*, **25**, pp. 21-85 (1941).



course an even greater buoyancy in the outgoing regenerators opposing the flow, but this is overcome by the pull of the stack resulting only in the existence of a large negative pressure between the checkers and the stack.

Just as in Example 1, the pressure against atmospheric can be plotted against position in the system. This is done in Figs. (5)4 c and d, which correspond respectively to the air and gas flows.

**5.1.5. Turbulent and Streamline Flow.** Osborne Reynolds showed that the flow of a fluid in a tube can be either turbulent or streamline according to whether the velocity is above or below a certain value. In streamline flow a filament of coloured material introduced into the flow persists for a great distance, whereas in turbulent flow it is rapidly mixed with the surrounding fluid by means of eddies. Reynolds was the first to introduce a dimensionless criterion, named after him, which is of great value in correlating the results of experiments on fluid flow in both the turbulent and non-turbulent regions. He showed that since viscosity and inertia are the only physical properties which affect the flow of a homogeneous incompressible fluid at constant temperature and possessing no free surfaces, then the ratio of the inertial to the viscous forces at any point in a given flow system uniquely determines the characteristics of the flow of such a fluid through the system. This ratio can be expressed as  $vD/\nu$  and is known as the Reynolds number  $R_e$ , where  $v$  is the mean velocity,  $\nu$  the kinematic viscosity and  $D$  a length characteristic of the system (e.g. pipe diameter). In furnaces gases behave almost as though they were incompressible as the pressure changes encountered are small compared with the absolute pressure, and hence Reynolds' condition for similarity can be applied except where significant changes in temperature occur within a region where they affect the flow pattern.

In the case of pipes which are perfectly similar geometrically (e.g. all smooth circular pipes of length great compared to the diameter can be treated as geometrically similar)  $R_e$  serves as a unique criterion of the flow conditions: for values of  $R_e$  below 2000 the flow is streamline, for values above 3000 it is turbulent.\*† In

\* This distinction between streamline and turbulent flow corresponds to the change over from one formula to another to represent the results of experiments on natural convection at different flow rates (see Chapter 4.3.1).

† It has since been shown that if the entry into the tube is made with sufficient care laminar flow can be obtained at values of  $R_e$  as high as 40,000. Such entry conditions do not occur in practical furnaces and the values given in the text are accordingly appropriate.



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addition to its use for determining whether the flow is streamline or turbulent, however, the Reynolds number is the appropriate dimensionless criterion for plotting against the friction factor in bends and changes of section, the friction factor being the dimensionless ratio of the total pressure drop to the dynamic pressure  $\frac{1}{2}\rho v^2$  at the point of entry. In the case of tubes, since the pressure drop must be proportional to the length and the dimensionless criterion involving the length is clearly the ratio of length to diameter  $L/D$ , the ratio of pressure drop to dynamic pressure  $\Delta p_r / \frac{1}{2}\rho v^2$  when divided by  $L/D$  will be a function solely of  $R_e$ . A single formula will thus apply to all smooth circular pipes.

In the cases of interest in high-temperature aerodynamics, the flow is nearly always turbulent, the Reynolds number being generally greater than 3000, but the criterion is still of great value for expressing the results of friction measurements in a way which enables them to be applied to all cases which are geometrically and dynamically similar (also as a criterion for the flow pattern studied in Sections 5.6 and 5.7). Appropriate formulæ for this type of calculation are given in Section 5.3.

It is possible to classify the type of flow in furnace systems, in spite of the fact that it is always turbulent, into steady flow and eddying flow. In steady turbulent flow any solid surface is surrounded by a boundary layer in which the flow is laminar, having a uniform gradient of velocity from zero at the solid surface to a value at the outer edge equal to that in the turbulent region. In eddying flow, on the other hand, the laminar "boundary layer" leaves the surface bounding the flow and relatively large eddies break off and pass into the wake of the surface. Such eddying flow is set up behind surfaces making a large angle with the direction of flow, e.g. sudden bends or changes in section. It is due to the occurrence of eddying flow that unless the expanding cone of a venturi has a total including angle of the order of  $15^\circ$  or less (see Section 5.3.2), this cone will have very little effect upon the overall pressure loss, which will therefore be as great as that occurring in an orifice of the same area; on the other hand, when the angle is less than  $6^\circ$  the boundary layer does not break away from the walls of the expanding cone and the dynamic head in the restriction is nearly all reconverted into static pressure.

A great deal of fundamental work on the nature of the eddies in the neighbourhood of obstructions has been carried out in wind tunnels in the last few years in connection with aircraft, but this



work has not yet been translated into terms suitable for use in furnaces. On the theoretical side turbulence is sometimes regarded as a condition of fluid flow where the instantaneous velocities at neighbouring points are connected only by a probability (and are not perfectly correlated, as they are in laminar flow) and some progress has been made along these lines. Possible applications of this work to the second problem of hot aerodynamics will be discussed in Sections 5.6–5.7.

## 5.2. Total Flow in a Circuit. Sources of Aeromotive Force

**5.2.1. Buoyancy.** As has been shown in Section 5.1.3, a column of hot gas will have a greater pressure relative to the surrounding atmosphere at the top than at the bottom if it is sealed off from flow. For example, a tube of hot gas closed at the top and open at the bottom will have atmospheric pressure at the bottom, but the pressure inside the tube will be greater than that of the atmosphere at the same height. This is because of the reduced density of the hot gas. This “buoyancy effect” provides the most important source of driving energy for the gases in a furnace system, a source which only a century ago was the only one available for all heating purposes, with the sole exception of the blacksmith’s bellows.\*

The calculation of the buoyancy head is a simple matter since it is given by the difference in weight of the columns of hot gases and cold atmosphere expressed in the appropriate pressure units. In calculating the buoyancy pressures for a given furnace system it is either possible to choose a fixed level, e.g. ground level, as a datum line, and calculate all buoyancy pressures above this level, or else the buoyancy pressure corresponding to each of the units into which the system is divided can be calculated and compared with the measured pressure change in that unit, in the manner indicated in Section 5.5. When the gas inside the system has the same density as

\* A very interesting use of the principle of the chimney occurs in the “Siemens syphon.” Siemens wished to supply producer gas to his furnace at a positive pressure without placing the producer a considerable distance below the furnace. He accordingly arranged a vertical brick stack rising from the outlet of the producer, carried the gases horizontally from the top of this stack through an iron tube in which they cooled appreciably, and then dropped them down to the same level as the producer in another brick shaft. Since the gases were considerably hotter in the rising than in the falling shaft they finished up with an appreciable positive pressure. With the introduction of injectors and fans, however, this syphon arrangement is no longer necessary.



### 5.2.1 THE SCIENCE OF FLAMES AND FURNACES

air at the same temperature, Fig. (5)5 can conveniently be used for buoyancy calculations. The pressure differential obtained from it must be multiplied by the height of the system to obtain the total buoyancy pressure.

Where the gases in the furnace have a density  $\rho_s$  appreciably different from  $\rho_a$ , that of air at the same temperature,  $\Delta p_B$  must be obtained from equation (5/5) i.e.:

$$\Delta p_B = 0.0147 \left( 1 - \frac{288}{273 + T_s} \cdot \frac{\rho_s}{\rho_a} \right) h$$

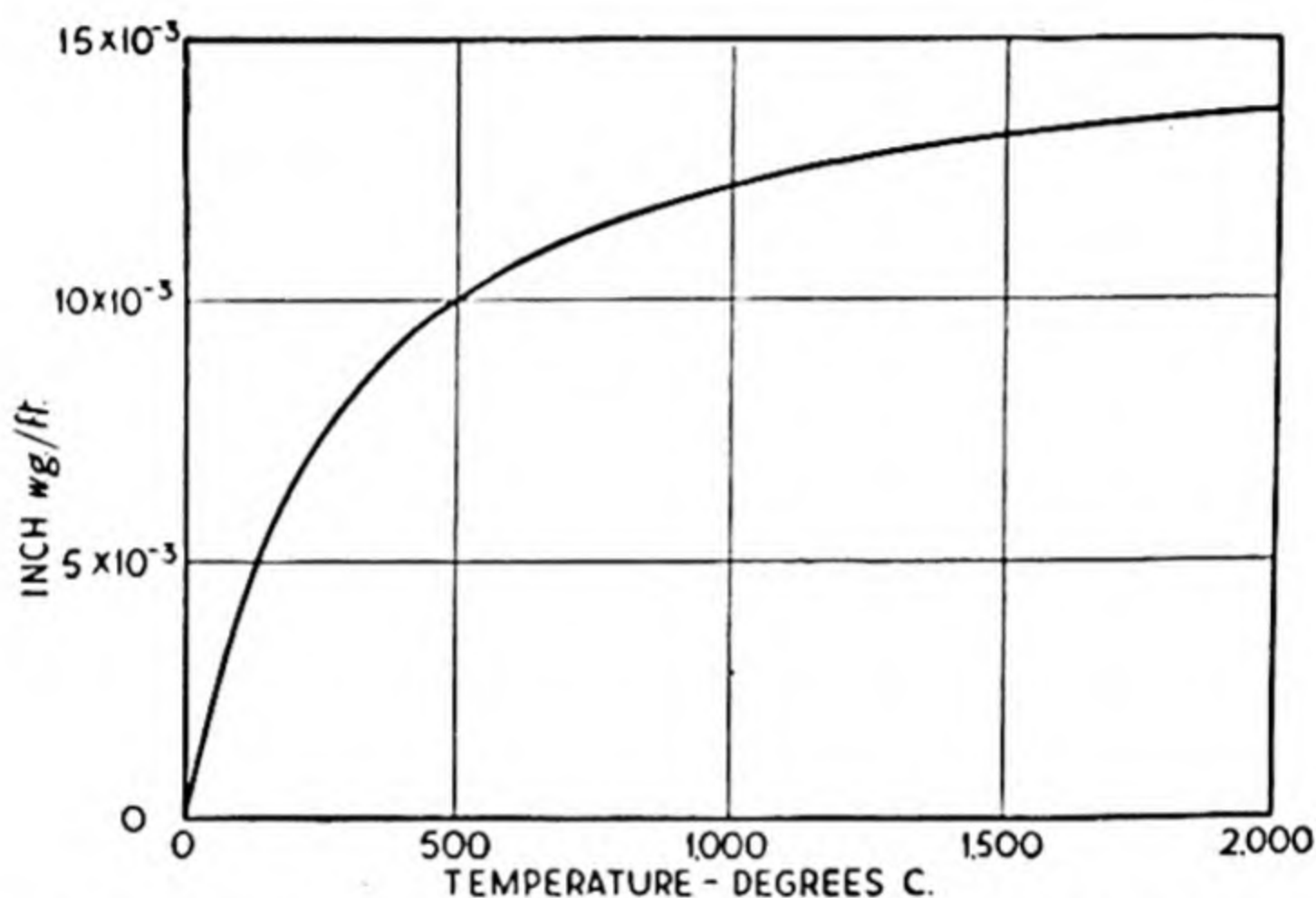


Fig. (5)5. Buoyancy Pressure developed by a Column of Hot Air one foot high when Surrounded by Air at 0° C., and 760 mm. (If the atmospheric temperature is above 0° C., then two pressures are found from the graph, one corresponding to the atmospheric temperature and one to the gas temperature. The difference between these is then the buoyancy pressure. For any other gas the pressure obtained must be multiplied by  $\rho_{\text{gas}}/\rho_{\text{air}}$ ).

where  $h$  = height of chimney in ft. ;

$T_s$  = temperature of hot gas in furnace in ° C. ;

$\rho_a$  = density of air at N.T.P. ; and

$\rho_s$  = density of gas at N.T.P.

The gases whose presence in large proportions most frequently cause this density ratio to deviate appreciably from unity are CO<sub>2</sub>, H<sub>2</sub>O and H<sub>2</sub>. In combustion gases the effects of the first two largely annul one another, but in producer gas or towns gas the effect of the third can be very great.

**5.2.2. Fans.** It can be seen from the formulæ of the preceding sub-section that the greatest possible draught obtainable with a 100-ft. chimney is about 1½ inches w.g. This corresponds to the dynamic head for cold air at a velocity rather less than 100 ft./sec.



Clearly, therefore, some other method of obtaining aeromotive force is necessary if velocities much greater than this are to be obtained or if it is desired to obtain a certain distribution of pressures in a system without great differences of level between one point and another. Fans and injectors are employed for this purpose.

When applied to furnace systems fans are usually intended to give a pressure difference of the order of 1-10 inches w.g. and can thus be of the radial flow type. The furnace designer does not usually design his own fans, but specifies to a maker the required volume of gas to be handled at a given pressure difference and temperature. It is thus not necessary here to enter into the question of fan design. A well designed fan can turn an appreciable fraction of the energy input into compression and kinetic energies.

Where higher pressures or suctions are required, the positive displacement blower is frequently used. This has a lower efficiency than a fan, but can operate to give high outlet pressures. The most modern development in this direction is the turbo-blower, which gives a very high efficiency even for high compression ratios.

**5.2.3. Injectors.** Although the injector is not aerodynamically simpler than the fan, it is useful to enter somewhat into the design of the former, since the furnace designer usually designs his own injectors because they are more individual to the job. The injector has no moving parts in the hot gas and hence is frequently used for moving gas at higher temperatures than would be possible with a fan. For example, a recent American innovation for the open-hearth furnace eliminates the reversing valves and stack and draws the waste gases out of the outgoing regenerators by an injector with a short funnel. One injector is connected to each end of the furnace and reversing the gas flow is done by changing from one injector to the other. Injectors are also frequently used where, as in the blast of a gas producer, it is necessary to mix two gas streams, one stream being supplied at high pressure and used to step up the pressure of the other. The principle of the injector can be discussed with reference to Fig. (5)6.\* The injecting fluid at a pressure of several lb./sq. inch is allowed to issue in a high-speed jet from the nozzle *D*. This jet passes through the throat *B* of a venturi, where the conversion of static to dynamic head creates a region of low pressure and hence draws with it the injected fluid from *E*. The dynamic head of the mixture is then reconverted into static head by

\* Alternatively, Kroll gives a set of curves which may be used (see ref. 5.2.)



### 5.2.3 THE SCIENCE OF FLAMES AND FURNACES

expansion in the cone  $BC$ . Kroll<sup>5.2</sup> has pointed out that the best form for the entrance to the throat is a well-rounded bell-mouthed entry, while a conical entry should have the angle  $\alpha > 20^\circ$ ; in this way the loss of momentum by friction is reduced to a minimum. The throat can be parallel and 5–10 times its diameter  $D_T$  in length, while the length  $X$  between the nozzle and the beginning of the throat should be adjustable so that the optimum position can be found by experiment. The diverging venturi section should have an angle  $\theta$  not more than  $10^\circ$  and a length of  $4\text{--}8 D_T$ . The nozzle can be a parallel hole without much loss of efficiency.

One method for calculating the dimensions of the throat and nozzle and the quantity of injecting fluid needed to produce a given gas flow against a given back pressure is the following.\* The laws of hydrodynamics of Section 5.1.3 are applied with the assumption

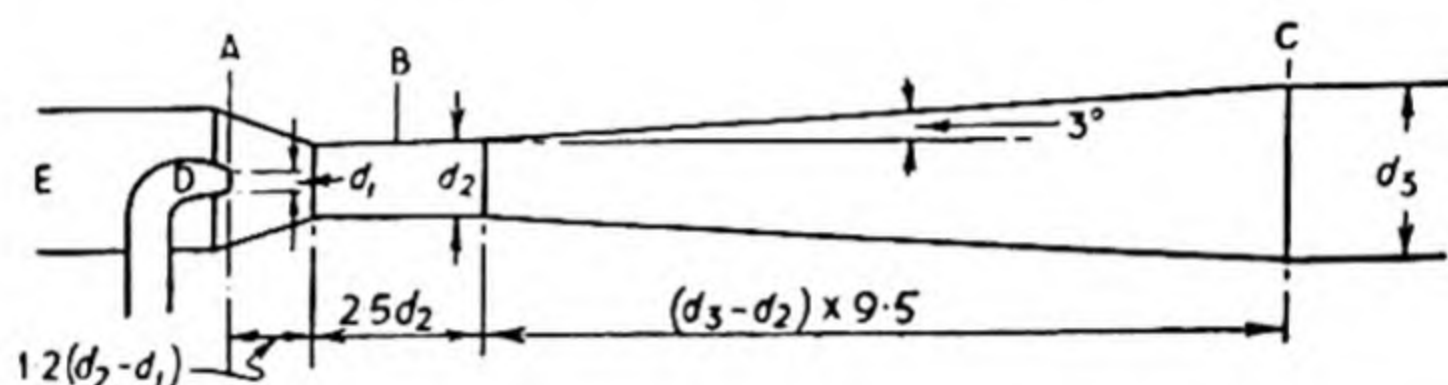


Fig. (5)6. Proportions of Venturi Injector.

(Courtesy of C. Smith and R. C. Aspland of Wellman Smith Owen Engineering Corporation, Limited.)

of certain losses due to friction. These losses may be higher when the injector is designed according to the above rules. The assumptions are :

(1) Momentum of inducing fluid at injector nozzle + momentum of induced fluid around injector nozzle =  $1.25 \times$  (momentum of mixture at injector throat). This means that 20% of the momentum is assumed to be destroyed by friction on mixing. The principle of conservation of momentum is applied here to the mixing of streams assumed to take place in a region with constant cross-sectional area for the reasons given in Section 5.1.3.

(2) Kinetic energy of mixture at throat =  $1.25 \times$  (pressure energy at end of expansion cone) + kinetic energy at end of expansion cone. This means that the reconversion of kinetic energy to

5.2 KROLL, A. E. : "The Design of Jet Pumps," *Chem. Eng. Progress*, 1, No. 2, p. 21 (Feb. 1947).

\* Quoted here by kind permission of the Wellman Smith Owen Engineering Corporation.



pressure energy in the expansion cone is assumed to be 80% efficient.

To use these assumptions we have the following formulæ :

The momentum per unit time of a quantity  $Q$  ft.<sup>3</sup>/sec. of fluid of density  $\rho$  lb./ft.<sup>3</sup> flowing in a tube of area  $A$  ft.<sup>2</sup>, is given by

$$\frac{\rho Q^2}{A} \text{ ft.-lb./sec.}^2 \quad (5/16)$$

while the kinetic energy carried by it past a fixed point per second is

$$\frac{\rho Q^3}{64 A^2} \text{ ft.-lb./sec.} \text{ or } \frac{\rho V^2}{64} Q \quad (5/17)$$

where  $V$  is the velocity of the gas-stream (ft./sec.).

The pressure energy of  $Q$  ft.<sup>3</sup>/sec. at a pressure of  $h_w$  inches w.g. is

$$5.2 \cdot h_w \cdot Q \text{ ft.-lb./sec.} \quad (5/18)$$

The calculation of an actual injector proceeds as follows. Suppose that the following are given :

- (1) the required pressure difference against which the injector has to operate ( $h$  inches w.g.) ;
- (2) the volume of gas  $Q$  ft.<sup>3</sup>/sec. ( $Q_o$  ft.<sup>3</sup>/sec. at N.T.P.) which is to be drawn into the injector ;
- (3) the temperature  $T$  of this gas ; and
- (4) the area of the inlet  $A_o$ .

Then the first step is to assume a value for the quantity of injecting gas  $Q'$  ft.<sup>3</sup>/sec. ( $Q'_o$  at N.T.P.),\* and for the velocity  $V$  of the mixture leaving the expansion tube at  $C$ . It is then required to calculate the area  $A_1$  of the throat of the injector and that of the jet  $A_2$ , together with the required pressure of injecting fluid. The temperature of the mixture of injecting and injected gases is calculated

\* This quantity would be taken from practical experience. Examples are :

(1) An injector chimney: in this case the quantity of air used would probably be about equal to that of waste gas injected.

(2) A gas producer injector: here the steam-air ratio is fixed by the required blast saturation temperature.

(3) A system for recycling flue gases: here the proportions would depend on the dilution required. In other cases it may be necessary to assume a value of  $Q'$  and see what this gives for  $\Delta p'$ . If then  $\Delta p'$ , the pressure required, is unduly high then recalculate with a larger value of  $Q'$  and conversely.



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from their temperatures and specific heats and the corresponding volume of the mixture at this temperature is deduced. From this the kinetic energy of the mixture where it leaves the expansion nozzle can be obtained using formula (5/16) and the assumed velocity  $V$ . The pressure energy corresponding to the difference in pressure between the gas entering the injector and that leaving at  $C$  is then calculated from (5/17). Using the second assumption it is then possible to calculate the kinetic energy of the mixture in the throat  $B$  and hence from formula (5/16) the area  $A_1$  of this throat.

The momentum of the mixture at the throat is then calculated from formula (5/15) and similarly the momentum of the injected gases at the inlet  $A_0$  can be obtained from this formula, since the area of the inlet is known. Then by applying assumption (1) the momentum of the inducing fluid at the injector nozzle  $D$  can be calculated. From this momentum the area  $A_2$  of this nozzle can be obtained, and the required pressure  $\Delta p'$  of injecting fluid is then given by the equation  $\frac{\rho'}{64} \cdot \frac{Q'^2}{A_2^2} = 5.2 \Delta p'$  (this is equivalent to the assumption that at the nozzle all the pressure energy is converted into kinetic energy).

A calculation of the injection performance of atmospheric gas burners by similar assumptions has been given by Westerdijk and Lantzius.<sup>5.3</sup> They deal with rich gas burners in which the gas issuing from a nozzle entrains air into a tube and this tube expands further along to pass the mixture through some type of head, such as a plate with a number of small holes, where it burns in the manner described in Section 3.2.3. for premixed gas flames. They apply Bernoulli's theorem (equation (5/6)) to give the drop in pressure of the air as it enters the mixing tube from zero velocity in the atmosphere; then the equation of momentum conservation (equation (5/9)) for the mixing tube with an efficiency factor of 50%, Bernoulli's equation to the expanding conical section of the diffuser, the momentum equation for the discontinuous transition to the ports and an empirical pressure coefficient for the burner itself are applied in turn. von Elbe and Grumer<sup>5.4</sup> have also calculated the air entrainment in gas burners using similar assumptions and have compared the results of the calculation with experimental results.

5.3 WESTERDIJK, T., and LANTZIUS: "A theory of Atmospheric Burners," *Engineering*, 168, p. 549 (25 Nov. 1949).

5.4 VON ELBE, G., and GRUMER, J.: "Air Entrainment in Gas Burners," *Ind. Eng. Chem.*, 40, p. 1123 (June 1948).



### 5.3. Total Flow in a Circuit. Resistances

In this section a brief summary is given of the various semi-empirical formulæ for the pressure drop  $\Delta p_R$  caused by friction when a known quantity of gas flows through various types of channel.

**5.3.1. Friction Loss in Long Channels.** The resistance to flow in channels where the length is large compared to the diameter is mainly due to the steady loss of energy by friction all along the walls, whereas the resistance in short restrictions is largely due to the dissipation of the increased dynamic head which cannot then be recovered in the re-expansion. The former type of resistance will be considered first. The general dimensionless equation for the friction loss in such channels is

$$\frac{\Delta p_R}{\rho v^2 / 2g} = 0.187 \cdot F \cdot L / D \quad . \quad . \quad . \quad (5/18)$$

where  $\Delta p_R$  is the pressure loss in inches w.g. ;

$L$  is the length of the pipe in ft. and  $D$  its internal diameter in inches ;

$\rho$  is the specific gravity of the gas at its actual temperature, relative to that of air at N.T.P. ;

$g$  is the acceleration due to gravity (32.2 ft./sec.<sup>2</sup>) ;

$v$  is the mean velocity of the gas in ft./sec. measured at its actual temperature  $T^\circ \text{C}$ . If  $Q_0$  is the quantity of gas flowing, measured in ft.<sup>3</sup>/sec. at N.T.P. and  $A$  is the area of the duct in ft.<sup>2</sup> then  $A$  is the cross-sectional area of channel in ft.<sup>2</sup>

$$v = \frac{Q_0}{A} \cdot \frac{273 + T}{273}$$

$F$  is the friction factor, which is a dimensionless function of the Reynolds number of the flow ( $R_e$ ) and of pipe roughness.

As it stands this equation represents neither the results of experiment nor of any detailed theory of the flow mechanism, but is purely the result of applying the assumption that Nature has no yardstick and hence cannot distinguish between a small system and a large one provided other factors are altered so as to compensate the effect of change of size. The quantity 0.187 is purely a conversion factor arising from the fact that the pressure drop  $\Delta p_R$  is expressed in inches w.g., while the dynamic pressure is expressed in other units involving the density of air at N.T.P.



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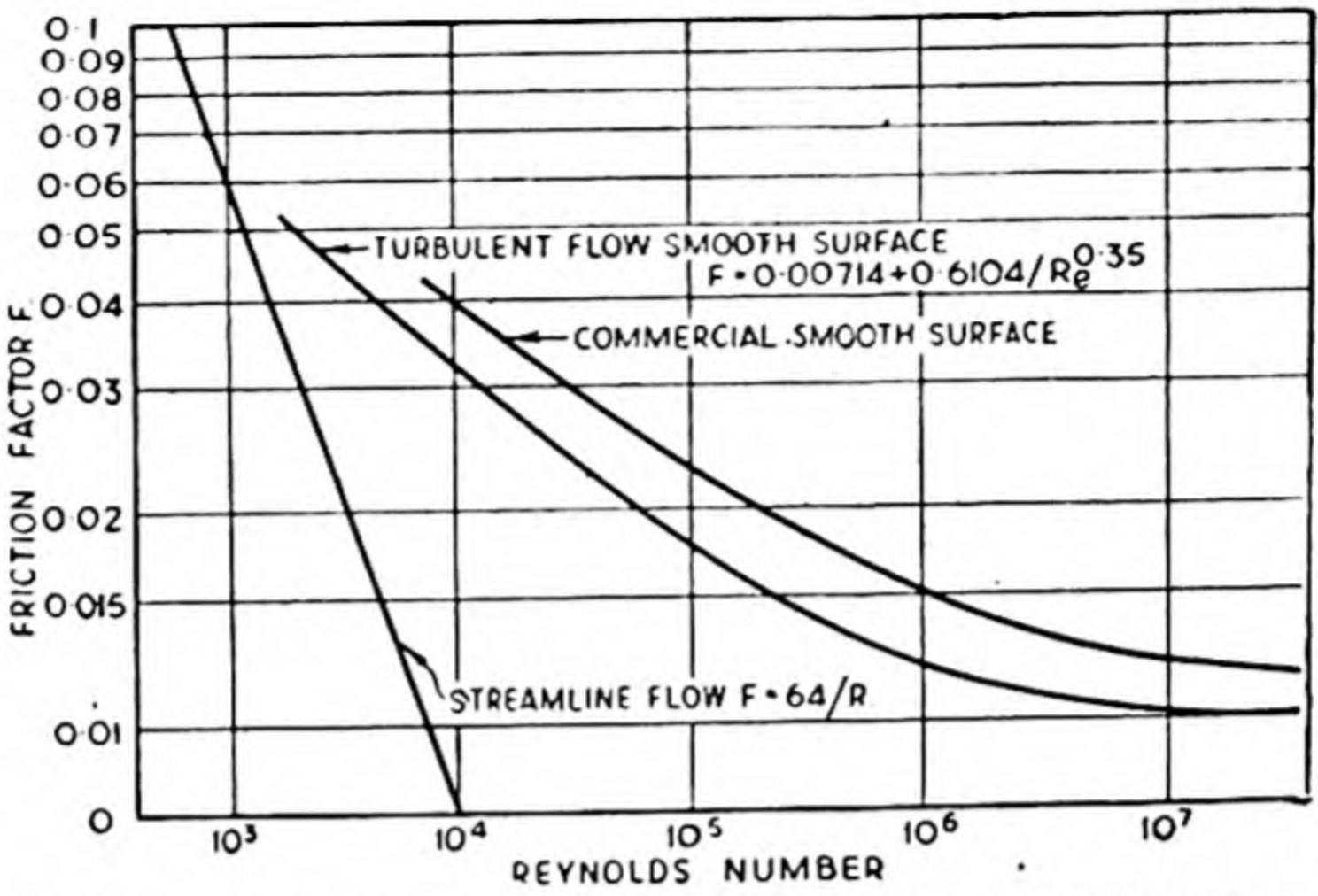


Fig. (5)7. Relation between Friction Factor and Reynolds Number for tubes.

The experimental results for the values of the friction factor  $F$  in terms of  $R_e$  and pipe roughness are summarised in Fig. (5)7, from which it will be seen that for large  $R_e$ \*  $F$  is independent of  $R_e$ , i.e. the pressure loss is proportional to  $V^2$ . This is a general characteristic of fully turbulent flow. In this region the following values may be used for  $F$ .

TABLE 5.3  
VALUES OF FRICTION FACTOR  $F$  FOR PIPES OF VARIOUS DEGREES OF ROUGHNESS

	Value of $D$ or $M$ , inches	Value of $F$ (clean and new)	After long use
Smooth drawn brass or steel	All values	0.008	—
Wrought iron, rivetted or welded	4	0.025	0.07
	20 or over	0.020	0.06
Cast " iron " " "	2	0.035	0.10
	4	0.028	0.09
" " " "	20	0.017	0.05
" " " "	4	0.028/0.038	0.09
Cement	40	0.013/0.018	0.05
" " " "	4	0.035	0.10
Brick	40	0.018	0.06
" " " "			

In this formula (5/18) the final term  $L/D$  takes account of the fact that the resistance to flow is due to the cumulative effect of the

\*  $R_e$  is given for circular pipes, by  $vD/12\nu$ , where  $D$  is in inches and  $\nu$  = kinematic viscosity of gas at  $T^\circ\text{C}$ . (see Tables 64 and 65 of *Technical Data on Fuel*) or for other pipes by

$$\frac{Q_o M}{\nu} \cdot \frac{273+T}{273}$$

where  $M=4/(\text{perimeter of channel in ft.})$ .



walls, since it implies, for example, that a pipe of twice the length has twice the resistance. In accordance with the principle of similarity  $L$  must appear in a dimensionless ratio and hence the ratio of its value to that of the other significant length (namely, the diameter) is inserted in the formula. In calculating the pressure drop in such systems as gas-producer mains, however, it should be noted that for a given quantity of gas flowing per hour, the velocity is inversely proportional to the square of the diameter and hence the pressure loss in fully turbulent flow is inversely proportional to the fifth power of the diameter. This means that a very small reduction of diameter due to the presence of deposits on the brick-work can produce an enormous increase in the pressure drop per unit length.

**5.3.2. Pressure Drop at Bends and Changes of Section.** Where the pressure drop is mainly due to the local turbulence and dissipation of energy caused by some violent alteration of the direction of flow, the pressure drop is best correlated with the average dynamic head alone. The formula for those cases is therefore :

$$\frac{\Delta p_R}{\rho v^2/2g} = 0.0156 \times K \quad \dots \quad (5/19)$$

where the symbols have the same meaning as in (5/10) and  $K$  is a quantity which is constant for fully turbulent flow such as occurs in furnace systems, but has to be evaluated by experiment for each shape.  $V$  by convention is taken as the velocity at the point of entry. Values of  $K$  for various cases are shown in Fig. (5)8 a, b, c, which apply respectively to bends, to changes of cross-sectional area and to certain other special cases. ✓

With violent changes of these types a great part of the dynamic head at the point of entry is converted into heat energy and the factor  $K$  is a measure of the amount so converted in terms of the dynamic head. Where  $K$  is greater than unity it is because the dynamic head is greater at some part of the obstruction where the stream narrows than the value at the point of entry which has been used in the formula for convenience. There is, however, one case in which dynamic head can be reconverted into static pressure (17 of Fig. (5)8b). This is where the gas is allowed to flow along a very gently expanding funnel with an included angle of  $6^\circ$  or less ; for slightly greater angles some recovery is still obtained, but with angles of  $30^\circ$  or  $45^\circ$  practically all the dynamic head in the narrow portion is lost. If the pressure balance is constructed using the total



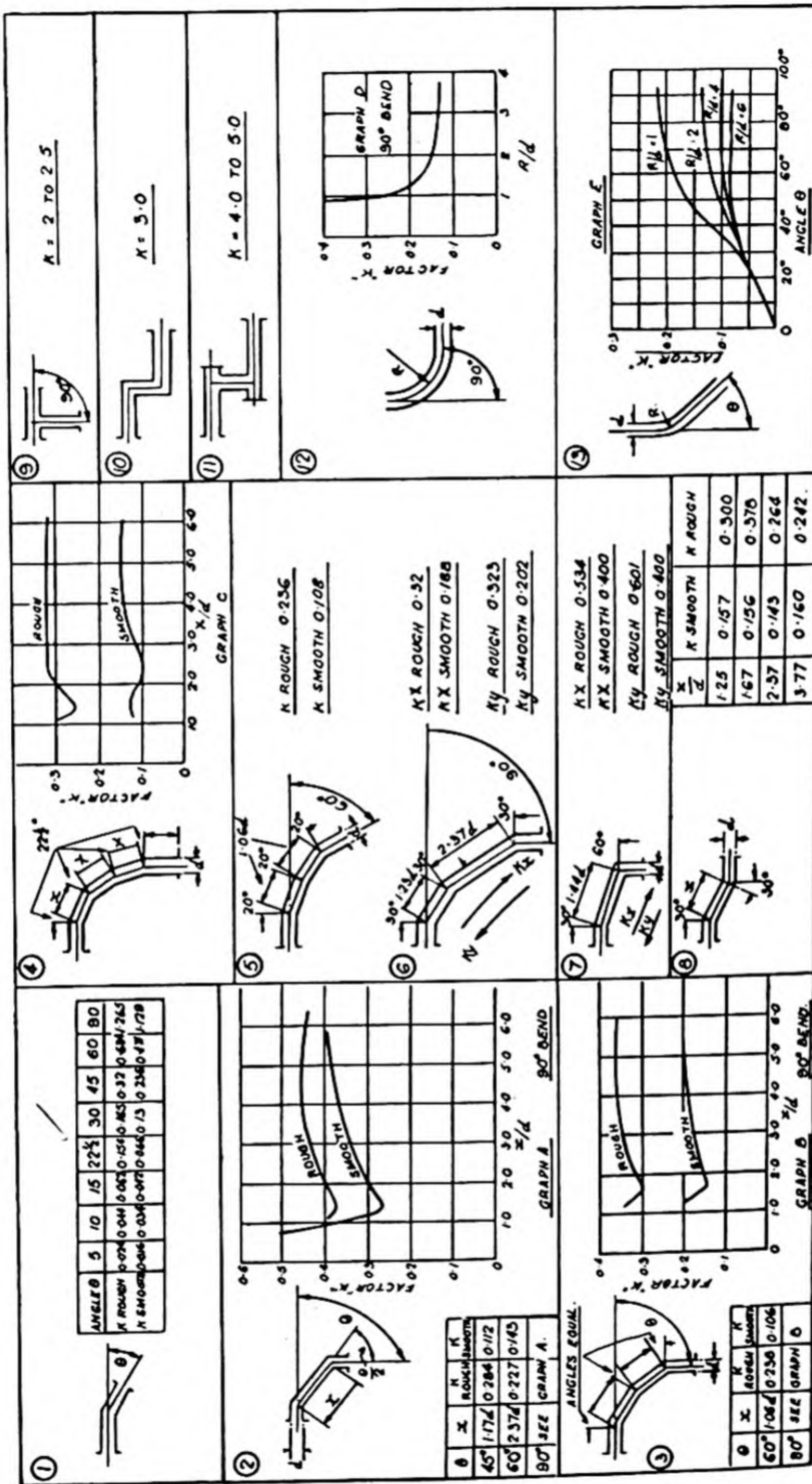
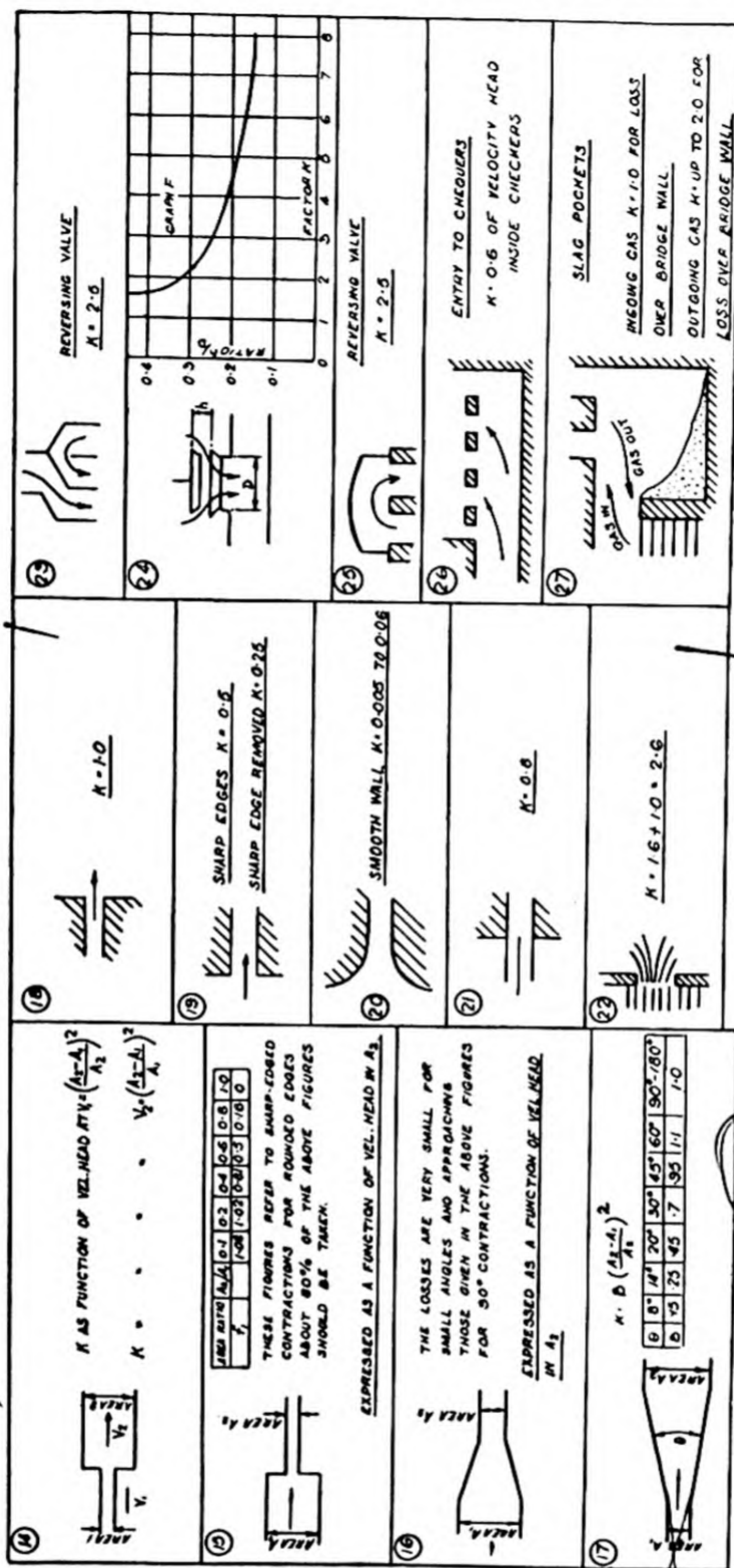


Fig. (5)8a. Friction factors for Different Shapes of Gas Conduit.

pressures throughout as discussed in 5.1.3 such recovery of dynamic head is automatically allowed for, and it is only necessary to regard the expansion as a resistance in which a certain resistance pressure is set up by the imperfections of the expansion. Where, on the





other hand, the pressure balance is constructed using the static pressure, the recovery of dynamic head must be allowed for by means of a velocity pressure change  $\Delta p_v$  given by (5/8) occurring in the expansion nozzle.



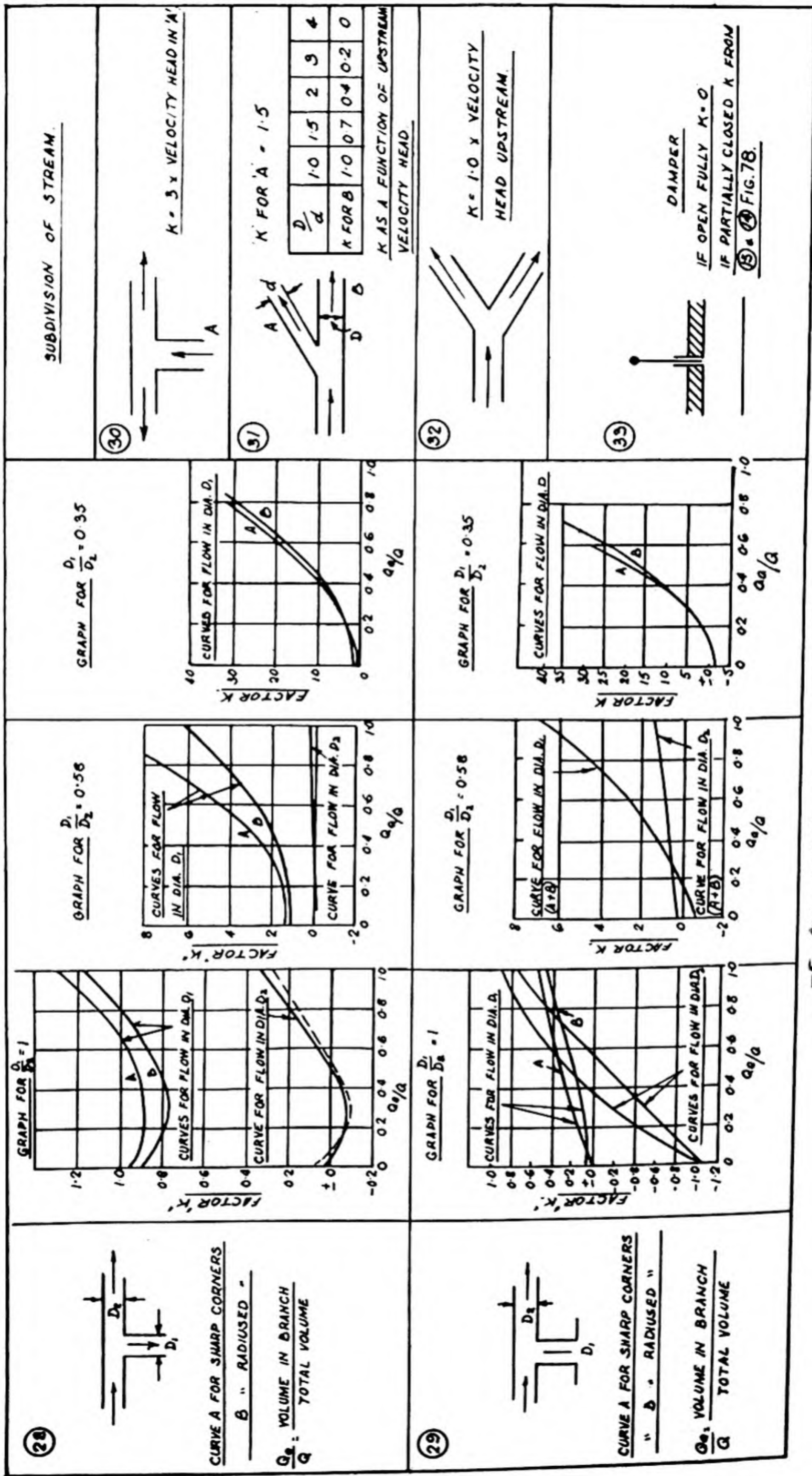


Fig. 58c. Friction Factors for Branching Flow.



The recovery of dynamic head, however, rarely plays a part in furnace design because it is not usually possible to allow sufficient length for a slow increase of area. The only two cases where such funnels do frequently occur are in Venturi meters and in injectors, and in each of these cases the expansion funnel can be treated along with the preceding contraction to give an overall resistance or aeromotive force which is calculated in the ways discussed in 5.4.2 and 5.2.3 respectively.

When bends and changes of section occur close together the combined resistance will not simply be the sum of the two, and further research is then needed to evaluate this combined resistance. In such a case the construction of air models can give results of the greatest value. It is a relatively easy matter to obtain in an air model of 1/12 scale Reynolds numbers of the same magnitude as in the original, because of the increase in kinematic viscosity of gases on heating. For example, if there is air in the furnace at 1000° C. the ratio of kinematic viscosities between furnace and model (air at 15° C.) is 12 : 1 and hence the velocities in the model need only be about the same as in the furnace. By constructing such a model, blowing a metered quantity of air through it and measuring the static or total head pressure change, the resistance can be readily determined, and, what is more important, complicated shapes and changes in the shape designed to alter the resistance can be tested in a matter of days, whereas changes in the furnace would take months and even then can rarely be compared reliably. A recent paper by Leys and Leigh<sup>5.5</sup> describes such measurements for the shapes concerned in the open-hearth system. Their model is illustrated in Fig. (5)19 (see Section 5.7.4). Such models are also of use in calibrating metering systems (see Section 5.4) where these cannot be constructed according to standard formulæ.

**5.3.3. Resistance of Beds of Solid Particles.** Formulæ for the resistance to flow of beds are subject to great inaccuracies for three reasons. In the first place the variation of packing from point to point has a very great effect on the resistance to flow, since if there are a few channels running right through they will clearly carry the great bulk of the gas. This effect is very rarely averaged out, so that formulæ which take account of the packing only by an overall voidage term are necessarily somewhat inaccurate. An example of

<sup>5.5</sup> LEYS, J. A., and LEIGH, E. T.: "Pressure and Flow Distribution in a Model of a Venturi-type Open-hearth Furnace," *J.I.S.I.* 165, p. 301 (1950).



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this effect is given by a case where the resistance of a coal bed was found<sup>5.6</sup> to be four times as great when the bed was packed by hand as compared with the value when it was filled by freely pouring coal, although the voidage was scarcely different.

The second factor which renders such formulæ inaccurate is the segregation of particles of different size. The effect of segregation is that while beds of uniformly sized particles can be expected to obey fairly accurate formulæ, when there is a large size distribution great errors can occur.

The third factor is the change of packing which occurs in a fuel bed when active combustion is taking place. The burning of the fuel bed keeps the particles separate and hence causes a greatly reduced resistance to flow.

As a result of all these factors, the resistance of a bed may vary over a range of about 10 or 20 to 1 and hence the extreme limits of resistance should be taken as being about four times greater and four times less than those given by any fixed formula. The one recommended here is that given by Carman<sup>5.7</sup>:

$$\frac{\Delta p_R}{\rho v^2} \cdot \frac{\epsilon^3}{LS_1} = 0.15 \left( \frac{\nu S}{v} \right) + 0.012 \left( \frac{\nu S}{v} \right)^{0.1} \quad . \quad (5/20)$$

where  $\epsilon$  = average porosity of whole bed (dimensionless) ;

$S$  = area of particle surface/unit volume of packed space (ft.<sup>-1</sup>) ;

$S_1 = S + 4/D$  (ft.<sup>-1</sup>), where  $D$  = diameter of container in ft. ;

$\nu$  = kinematic viscosity of fluid in ft.<sup>2</sup>/sec. ;

$v$  = actual velocity of fluid in empty container, ft./sec. ;

$L$  = thickness of bed in ft. ;

$\rho$  = density of fluid in lb./ft.<sup>3</sup> ; and

$\Delta p_R$  = pressure drop across in inches w.g.

The left-hand side of this formula expresses the ratio of the pressure drop to the appropriate dynamic head, multiplied by a factor ( $\epsilon^3$ ) which represents an attempt to allow for the effect of varying packing as assessed by the voidage ( $\epsilon$ ) and divided by the factor  $LS_1$ , which is equivalent to the term  $L/D$  in the formula for long pipes, since  $S_1$  is equivalent to the reciprocal of a certain type of average particle size corrected for wall effects. The right-hand

<sup>5.6</sup> Joint Report of the B.I.S.F. and B.C.U.R.A. : *The Flow of Air in a Gas Producer* (June 1939).

<sup>5.7</sup> CARMAN, R. C. : *Trans. Inst. Chem. Eng.*, **15**, p. 150 (1937).



side is a function of the Reynolds number ( $R_e = v/\nu S$ ), which is proportional to  $1/R_e$  for small values of  $R_e$ , but which, unlike the corresponding function for pipes, never reaches a steady value for large  $R_e$ . This is because even at high rates of flow the flow always remains stream-line in some of the smaller channels.

**5.3.4. Resistance of Regenerators.** For fairly clean regenerators working at pressures close to atmospheric Kistner<sup>5.8</sup> gives the following formula :

$$\frac{\Delta p_R}{\rho_0 v_0^2} = \frac{C.T.H.}{d^{0.25}} \text{ inches w.g.} \quad . \quad . \quad . \quad (5/21)$$

where  $v_0$  is the velocity through the free section of one course of the checkers reduced to N.T.P. ;

$\rho_0$  is the density of the gas at N.T.P. in lb./ft.<sup>3</sup> ;

$T$  is the mean temperature of the gas in the regenerator in ° K. ;

$d$  is the side of the square flues in ft. (in the case of flues other than square,  $d=4$  times the area of one flue in sq. ft. divided by the perimeter in ft.) ;

$H$  is the height of the regenerator in ft. ; and

$C$  has the value  $5.1 \times 10^{-6}$  in the case of straight-through checkers and  $7.1 \times 10^{-6}$  in the case of double-staggered checkers.

This formula is based on actual experiments on a regenerator of standard dimensions and hence can be relied upon provided the packings are more or less standard and the flues are not unduly blocked by deposits. It is not, however, in the fully dimensionless form required by the theory of similarity and hence cannot be extended much beyond the dimensions and proportions of a normal checker system.

**5.3.5. Resistance of the Furnace Walls to Leakage.** In the case of openings and cracks of known dimensions, the resistance can usually be calculated by treating them as orifices (when the aperture is large compared to the wall thickness) or flat channels (when the aperture is narrow compared to the wall thickness, the more usual case). In the case of orifices, the calculation is made as in diagram 22 of Fig. (5)8 (Section 5.3.2), while in the case of flat channels the formula (5/18) of Section 5.3.1 is used, the Reynolds number being calculated from the equivalent diameter as indicated in the footnote.

<sup>5.8</sup> KISTNER, H. : *Archiv. f. d. Eisenhüttenwes.*, pp. 751-68 (1929-30).



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As regards the flow through the bricks themselves, Bansen<sup>5.9</sup> has given a formula :

$$Q = \frac{\alpha A \Delta p}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5/22)$$

where  $Q$  is the rate of gas leakage in ft.<sup>3</sup>/sec. ;

$A$  is the area of wall in ft.<sup>2</sup> ;

$\Delta p$  the pressure difference in inches w.g. between the two sides ;

$d$  is the thickness of the wall in inches ; and

$\alpha$  is a constant which depends very much on the condition of the wall, varying between 0.16 for bricks laid with good mortar joints, to 5.6 for bricks laid dry without special care.

Owing to the large variation in the values of  $\alpha$  this formula can only be used to give a very rough idea of the leakage, which will clearly depend on the whole state of the wall. By enclosing the brickwork in steel casing, broken, if necessary, with sand-sealed expansion joints, such leakage can be almost entirely eliminated.

## 5.4. Measurement of the Total Quantity of Gas Flowing in a Circuit

**5.4.1. Measurement by the Pressure Differential across a Restriction. Theoretical Basis.** The simplest method of measuring the total quantity of gas flowing in a single channel flow system is to place a restriction in the channel and measure the pressure differential set up by the gas as it flows through this restriction. At a point  $S_1$  inside the restriction the gas has a velocity ( $v_1$ ) higher than its velocity ( $v_2$ ) at a point  $S_2$  in the main channel, consequently the static pressure  $p_1$  at  $S_1$  is, by Bernoulli's theorem (Section 5.1.3, formula (5/6)) lower than the static pressure  $p_2$  at  $S_2$ . Since the static pressure can be readily measured by means of a tapping or set of tappings flush with the wall, a single measurement of the difference of pressure between two such sets of tappings can be made to give a knowledge of the total quantity of gas flowing. In practice the pressure difference cannot be equated to the difference in dynamic head at the two points because of the existence of eddies and non-uniform velocities. Hence, the principle of such methods of measurements is to calculate from Bernoulli's theorem the basic

<sup>5.9</sup> BANSEN : "Gasdurchlässigkeit v. Silikastein, Mörtelfugen und Mauerwerk im Siemens-Martin-Ofen," *Archiv. f. d. Eisenhüttenw.*, 1, pp. 687-92 (May 1927-28).



formulae and then insert into the formulae empirical coefficients obtained from measurements on geometrically similar systems. Such coefficients are accordingly functions of the similarity criteria, particularly the Reynolds number, and may be determined directly or by the use of models.

If therefore Bernoulli's theorem is applied to the case where there are no buoyancy effects (either a horizontal channel or else gas of the same density as the atmosphere) then it gives, if the pressures are measured in lb./ft. sec.<sup>2</sup> so that no conversion factors are needed:

$$p_1 + \frac{\rho v_1^2}{2} = p_2 + \frac{\rho v_2^2}{2}$$

hence

$$p_2 - p_1 = \frac{\rho}{2}(v_1^2 - v_2^2)$$

where  $p_1$  is the static pressure in the restriction,  $S_1$  and  $p_2$  the static pressure at the point of low velocity  $S_2$ , and  $v_1, v_2$  are the corresponding velocities. Now the total quantity of gas flowing is the same in each place, hence

$$Q = v_1 \cdot A_1 = v_2 \cdot A_2$$

also, let  $m = A_1/A_2$  ( $m$  is known as the area ratio; when both the orifice and the pipe are round,  $m = d_1^2/d_2^2$ ):

$$p_2 - p_1 = \frac{\rho Q^2}{2} \cdot \frac{1 - m^2}{A_1^2}$$

or

$$p_2 - p_1 = \frac{\rho Q^2}{2} \cdot \frac{1 - m^2}{m^2 A_2^2} \quad \dots \quad (5/23)$$

If the pressure difference is measured as a height  $h$  of a fluid of density  $\rho_1$ , then we get:

$$Q = \sqrt{\left(2gh \frac{\rho_1}{\rho}\right)} \cdot \frac{A_1}{\sqrt{(1 - m^2)}}$$

or

$$Q = \sqrt{\left(2gh \frac{\rho_1}{\rho}\right)} \cdot \frac{A_2 \cdot m}{\sqrt{(1 - m^2)}} \quad \dots \quad (5/24)$$

Each form of this formula has value for one type of calculation so that they are both included.



### 5.4.1 THE SCIENCE OF FLAMES AND FURNACES

In practice frictional and geometrical factors are allowed for by multiplying the right-hand side of the formula by the product  $C \cdot Z$ , where  $C$  is a coefficient of discharge which for each type of restriction depends only on the area ratio  $m$ .  $Z$  is the product of three factors, one of which allows for variation of the coefficient with the Reynolds number  $R_e$  in the restriction; the second is a correction factor for the fact that the volume of the gas changes with pressure; and the third is a roughness factor which allows for the fact that commercial pipes have roughnesses of about the same magnitude whatever the diameter, so that the smaller the pipe the greater the correction. The working formulæ given in Section 5.4.2 are the same as these formulæ with coefficients inserted to take account of the fact that the dimensions of the quantities which are the most convenient expressions of  $Q$ ,  $p$ ,  $A$ , etc., are not self-consistent.

There are two main types of flow measurement in pipes based on restrictions, the Venturi and the Orifice. The venturi is a fully stream-lined restriction in which the channel is tapered down to the desired smaller bore and tapered back again to the original bore with the tapers sufficiently gentle to avoid the setting up of eddies (cf. Section 5.3.2). With such a system the flow at any position fills the whole cross-section of the pipe. Hence the pressure differential set up by the venturi will always correspond to the difference of static pressures caused by the difference in velocities near the wall. For fully turbulent flow the velocity is nearly constant across the flow and hence in this case one would expect the pressure difference calculated from Bernoulli's theorem with the simplifying assumptions used above to be correct; hence that  $C$  should be unity. In actual fact  $C$  is almost exactly 0.99 at all values of  $m$ , while the correction factor for  $R_e$  is 0.95 when  $R_e = 5000$  and  $m$  small, rising to 1 for large  $R_e$  and  $m$ . In the venturi meter one tapping is situated just before the reduction of diameter commences and the other halfway along the narrowest part.

The orifice consists of a restriction formed by a thin plate with a circular hole. The tappings in this case are both situated in parts of the channel having the full diameter. The two most usual arrangements are (1) that in which the tappings are 1 diameter  $D$  upstream and  $\frac{1}{2} D$  downstream, from the plate, called the  $D$ ,  $D/2$  tapping, and (2) the corner tapping system in which the tappings are close to each side of the orifice plate. The former is often more convenient. The downstream tapping has a pressure corresponding



approximately to the static pressure in the restriction because, owing to the formation of a turbulent jet after the orifice, the flow does not immediately fill the tube but retains a high velocity for a distance equal to several pipe diameters. As one would expect, in this case the pressure difference is less than for the true measurement of the static pressures at the low and high velocities and hence  $C$  is lower, having a value around 0.6.

The venturi has to be constructed accurately throughout the whole length of the composite restriction, whereas in the case of the orifice the orifice itself is the only part which needs to be constructed with high accuracy; hence the venturi is more expensive. On the other hand, in the latter the greater part of the dynamic head is reconverted into static head and hence there is a very small overall resistance to the fluid flow (of the order of one-tenth of the measured pressure difference). The venturi is therefore particularly useful for metering gases where it is essential to have a minimum restriction to the flow. Venturi meters can be made of sheet metal rolled and welded or, on the smaller scale, they can be turned from solid metal. Recently, venturi meters constructed in refractory concrete have been used successfully to measure hot raw producer gas. Vertical venturi throats<sup>5.10</sup> have the advantage that the deposits can be periodically blown down by air or steam jets into some place where they do not interfere with the area of the throat, but Cameron<sup>5.11</sup> has succeeded in using a horizontal venturi by having four radial blowing pipes just upstream of the throat.

Orifice meters can be used on hot, dirty gas and the measuring parts can be constructed of heat-resisting steel<sup>5.12</sup> or even brick. The permanent loss of head depends upon  $m$ , lying between the whole of the pressure differential ( $m=0$ ) and one-third of it ( $m=0.7$ ). This means that the metering process involves an increase in the power required in the case of fan-operated systems or a reduction in the maximum quantity of gas that can be drawn through in the case of chimney-operated systems. Where orifices are used on hot gas difficulties arise similar to those with venturi meters, due to the presence of dust and tar which cause wear on the plate and deposits on the upstream face.

5.10 *I.S.I. Special Report* No. 37, Section II, Pt. 3 (1946).

5.11 CAMERON, W. G.: "A Note on the Metering of Crude Producer Gas in a Horizontal Main," *J.I.S.I.*, **162**, p. 185 (June 1949).

5.12 SINCLAIR KERR, J., MORGAN, J. V., GOLDSBROUGH, W., and BINNIE, D.: *I.S.I. Special Report*, **22**, pp. 65-81 (1938).



**5.4.2. Calculation of Orifices and Venturis.\*** (a) *Calculation of quantity of gas flowing for given orifice dimensions and pressure difference (h inches of water).* When the orifice and pipe are hot it is necessary first to calculate their true working diameters ( $d$  and  $D$  respectively), i.e. including thermal expansion, by the method of successive approximation.  $m$  is then calculated as  $d^2/D^2$  and the value of  $C$  is obtained from data sheet 7 or 8 of the B.S.I. Code referred to above, according to whether an orifice or a venturi is being used. The quantity  $Q$   $n$  ft.<sup>3</sup>/hr. of gas (including water vapour) is then given approximately by

$$Q = 7440 \times \frac{C \cdot d^2}{\sqrt{(1-m^2)}} \cdot \sqrt{h} \cdot \sqrt{\left(\frac{p_1}{1.8T\delta}\right)} \quad (5/25)$$

where  $p_1$  is the absolute pressure of the gas in lb./in.<sup>2</sup> measured at the high-pressure tapping ;

$\delta$  is the specific gravity of the gas (including water vapour) relative to dry air at the same temperature and pressure ;

$T$  = absolute temperature of the gas, ° K. ;  $m = A_1/A_2$ .

This approximate flow is then used to calculate the Reynolds number,  $R_e$ , in the orifice from the formula :

$$R_e = \frac{Q \cdot \delta}{211 \cdot d \cdot \eta} \quad (5/26)$$

where  $\eta$  is the absolute viscosity of the gas at the working conditions in c.g.s. units (*Technical Data on Fuel*, p. 54, Table 68). It can also be calculated from table 4.1 (col. 4) p. 182, by multiplying by the density.

The corrections for the Reynolds number,  $R_e$ , and for  $D$  are obtained from the same data sheet, corresponding to the type of metering device to be used, while the correction for expansion is obtained from data sheet 11. The factor  $Z$  is then defined as the product of these three corrections and the flow is recalculated with greater accuracy by multiplying the previously calculated flow by  $Z$ . In most cases in connection with furnace work the tolerance on the resulting figure for  $Q$  is well within the limits of general accuracy, which are usually  $\pm 3\%$  or  $4\%$ . It is necessary, however, when irregularities in the channel occur fairly close to the orifice to make

\* For fuller details of orifice and venturi flow measurement and for the figures and graphs see B.S.I. Code BS.1042, 1945 : "Flow Measurement." The references in this section to orifices refer to the " $D$ ,  $D/2$ " tappings throughout.



sure that they do not come closer than the minima laid down in data sheet 3. In most cases for orifices, fifteen diameters of straight pipe upstream and five diameters downstream should give satisfactory accuracy. For cases where these conditions cannot be filled see below, Section 5.4.3.

(b) *Calculation of the diameter of the orifice necessary to give a desired pressure difference for a known flow.* This case occurs more frequently in furnace investigations where the gas flow is roughly known and it is desired to insert a restriction into a pipe and measure accurately the flow. A rough estimate  $Q_1$  of the quantity of gas is made from the fuel consumption and gas analyses, and the appropriate orifice is calculated to give a chosen pressure differential  $h$  at this flow  $Q_1$ . It is then inserted and the actual differential is measured; from this measurement the true value of  $Q$  is obtained. The appropriate orifice is calculated from  $h + Q$  from the formula:

$$\frac{Cm}{\sqrt{(1-m^2)}} = \frac{Q_1}{7440 \times D^2 \times \sqrt{h} \sqrt{\left(\frac{p_1}{1.8T\delta}\right)}} \quad (5/27)$$

The right-hand side of this formula is calculated from the known values. The value of  $m$  corresponding to the calculated value of

$\frac{Cm}{\sqrt{(1-m^2)}}$  is read off from data sheet 7 or 8 and the uncorrected value

of  $C$  is also obtained from this figure. The calculated value of  $\frac{Cm}{\sqrt{(1-m^2)}}$

is divided by  $C$  to give  $\frac{m}{\sqrt{(1-m^2)}}$  and from this  $m$  and hence

$d/D$  and finally  $d$  are calculated. This enables the three correction multipliers to be calculated as before to give  $Z$ ; then the previously

calculated value of  $\frac{m}{\sqrt{(1-m^2)}}$  is corrected by dividing it by  $Z$ ,

and thus revised values of  $m$  and of  $d$  are obtained. It is necessary to proceed in this manner because the corrections for  $R_e$ , etc., cannot be deduced until an approximate value of  $d$  is known.

**5.4.3. Special Uses of Orifices and Restrictions.** When the necessary lengths of reasonably smooth straight pipe are available, the standard orifices can be used without calibration. When, however, the necessary length of straight pipe is not available, it is still possible to use the orifice principle provided a calibration is carried out, since the pressure difference across the orifice will still



### 5.4.3 THE SCIENCE OF FLAMES AND FURNACES

be a function of the quantity flowing, although the coefficient will differ from the standard one. Such a calibration is most simply carried out by measuring the same gas flow by a separate method somewhere else along the line, e.g. measuring the flow velocity at all points across the tube, as discussed below in Section 5.4.4. Hot gas systems can be calibrated with cold air, although probably not over all of their normal working range. Since the kinematic viscosity goes up about ten times in going from room temperature up to the temperature concerned in hot gas flow measurements, the velocities of the air in such a calibration need only be about one-tenth of the hot gas velocities, and hence such a calibration is in general fairly easy. The pressure differentials, however, must be measured with more sensitive equipment on the model as they will be less than those obtained with the hot gas at corresponding Reynolds numbers. This is usually quite feasible, however, as the model tests are carried out in the laboratory where more delicate apparatus is permissible compared with the works. Finally, it is possible to construct a small-scale model and obtain the correct Reynolds number by the use of a sufficiently high velocity or at lower velocities by using water, since water has a much lower kinematic viscosity than gas (when both are at atmospheric temperature, say  $15^{\circ}\text{C.}$ ,  $\nu_{\text{water}} = 0.011 \text{ c.g.s.u.}$ ;  $\nu_{\text{air}} = 0.145 \text{ c.g.s.u.}$ ). In using small-scale models care has to be taken to ensure that the geometrical similarity applies to the system as it will actually be in practice and not just as it operates when first built, e.g. allowance must be made for erosion of refractories and build-up of dust and tar. If these requirements are satisfied, restrictions for producing differential pressures which deviate very considerably from the standard can be used with a fair degree of accuracy to meter the gases in furnace systems.

When using orifice meters on hot-gas systems, care must be taken to have the pressure tappings at exactly the same height in the system and to run the connecting tubes horizontally to a point where the gas in them is cool, or else to make allowances for buoyancy effects by calculation. An example of a case where one tapping had to be higher than the other so that such a calculated correction was essential occurs in the paper by Gooding and Thring,<sup>5.1</sup> where the "tongue" in each port of a glass tank furnace was used as a restriction to measure the flow through this port relative to that through the other ports.

<sup>5.1</sup> GOODING, E. J., and THRING, M. W.: *Trans. Soc. Glass Tech.*, **25**, pp. 21-85, (1941).



It is also possible to use the orifice principle in some cases where the orifice connects two chambers so large that the velocity in them is negligible. In this case the orifice can be calculated in the usual way,  $m$  being taken as zero. Tappings are taken in walls at right angles to the partition wall, one on each side, and they should be fairly close to the partition wall. Where the "chamber" on the inlet side is the outside atmosphere, the upstream tapping can be omitted if the open side of the manometer is close by the orifice, and both orifice and manometer are sheltered from draughts. In the case of flow through an orifice at the end of a pipe into an open space, the value of  $m$  used is the value corresponding to  $D$ , the diameter of the pipe on the inlet, namely  $d^2/D^2$ , provided this pipe is large enough to give a value of  $m$  less than 0.15. If this condition cannot be satisfied model tests are necessary. In the case of flow from an open space into a pipe,  $m$  is taken as zero provided the area of the pipe on the downstream side is not less than seven times that of the orifice. In these two latter cases the tapping in the pipe should be situated as for an orifice, viz. at distances  $D$  upstream or  $D/2$  downstream respectively. If these conditions are satisfied orifices can give accurate results in spite of the deviation from the standard geometrical arrangements due to the absence of a pipe on both sides, so that separate calibration is unnecessary.

**5.4.4. Measurement of Quantity by Measurement of Velocity at Points on a Cross-section.** If the cross-section of a channel in which the flow is parallel is subdivided for argument's sake into a large number of smaller areas and the velocity at the centre of each of these is measured, the total quantity of gas flowing along the channel can be estimated by summing the products of the areas and the corresponding velocities. ~~If the areas are made equal then it is only necessary to multiply the total area by the arithmetic mean of the velocity measurements.~~ Unlike the orifice or venturi meters, such a method of measuring does not cause any appreciable resistance to the stream, but on the other hand it is necessarily laborious.

In the case of rectangular channels the area can conveniently be subdivided into a set of geometrically similar smaller rectangles of equal size, which should not be less than 16 in number. For good accuracy, when the number of rectangles is less than 10 three readings should be taken and averaged for each rectangle next to the wall, and five for those in the corners. In the case of circular channels the area is best subdivided into a number of concentric



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rings of equal area and the velocity measurements made at the points of intersection of these rings with two perpendicular diameters. Ten such rings are recommended. The points for the measurement have accordingly the distances given in Table 5.4.

TABLE 5.4  
DISTANCES ALONG THE AXIS CORRESPONDING TO ANNULI  
OF EQUAL AREAS FOR CIRCULAR PIPES OF DIAMETER  $D$

$0.026D$	$0.658D$
$0.082D$	$0.774D$
$0.146D$	$0.854D$
$0.226D$	$0.918D$
$0.342D$	$0.974D$

An alternative method is to take measurements at roughly equal intervals along the diameter ; a smooth curve is then plotted from them and the appropriate velocities for the distances of Table 5.4 measured off from the curve. In cases where the pipe contains deposits, the area must be modified accordingly.

An alternative way of using a velocity-measuring device to deduce the quantity of gas flowing in a duct is as follows. The velocity is measured at a fixed point in the cross-section with a permanently installed velocity meter and the relation between this velocity and the total flow quantity or the mean velocity is ascertained by a series of experiments covering the range of Reynolds numbers which will occur in practice, by a temporary method of measuring the total flow quantity. The total quantity flowing in the channel can subsequently be deduced from this velocity. The calibration is usually carried out by means of the method just described of traversing the velocity-measuring device across the whole channel, or an orifice or venturi can be temporarily installed. This calibration can, however, be avoided in the case of straight circular ducts by fixing the velocity-measuring device on the axis of the pipe and using Fig. (5)9. This figure gives the ratio of the mean velocity to that which would be measured on the axis as a function of the Reynolds number. The form of velocity distribution in a cross-section of a pipe is, however, so greatly dependent upon the length of straight pipe upstream of the cross-section that adequate length to apply Fig. (5)9 is seldom available and this method of measurement is rarely possible in practice.

The simplest method of measuring velocity is by means of the *Pitot-static Tube*, which, like the orifice meter, is a practical appli-



cation of Bernoulli's theorem. The pitot-static tube is an instrument for measuring the difference between the static pressure and the total pressure at any given point in a flowing gas ; the difference between these is, by Bernoulli's theorem,  $\frac{1}{2}\rho v^2$ , where  $v$  is the velocity at this point. By suitable design of the pitot-static tube both the static and total heads can be measured accurately and hence the coefficient for the tube can be made close to unity. In this case the velocity  $v$  is given by the formula :

$$v = \frac{18.29}{\sqrt{\rho}} \cdot \sqrt{h} \text{ ft. sec.} \quad \dots \dots (5/28)$$

where  $h$  is the measured pressure difference expressed in inches w.g.

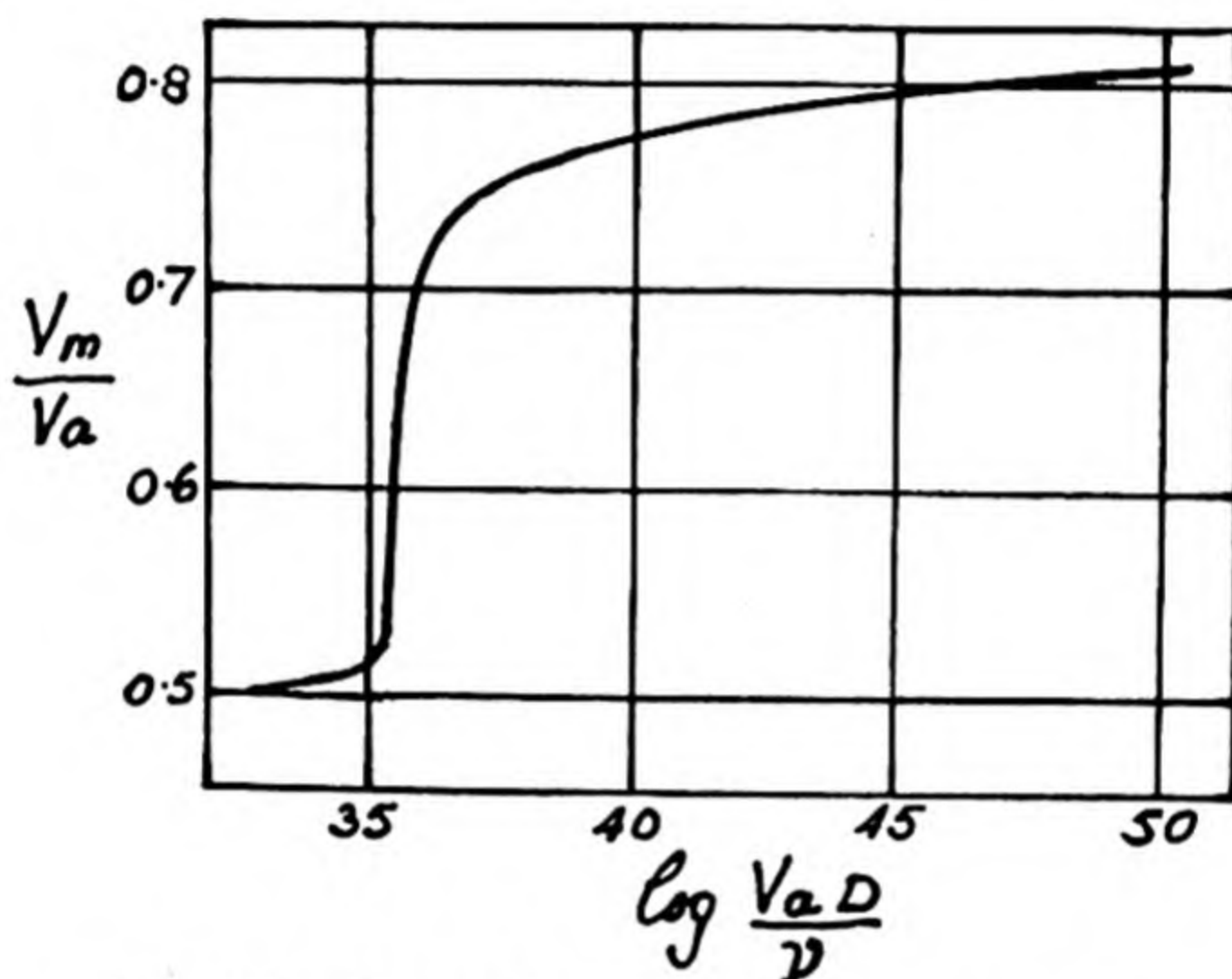


Fig. (5)9.  $V_m/V_a$  in Circular Pipes.  
 $V_m$  = Mean velocity;  $V_a$  = Axial velocity).

and  $\rho$  is in lb./cu.ft. at a temperature of 16° C. and a pressure of 1 atmosphere.

It is possible to have the tube for measuring the static pressure mounted flush in the wall parallel to the gas flow, while the total head tube or pitot tube consists simply of a tube facing directly into the gas stream. This arrangement is particularly suitable where the dimensions of the pressure tapings must be very small to avoid disturbing the flow, a condition which is not often met in furnace systems. Alternatively, the combined pitot-static tube can be used. For furnace work the N.P.L. hemispherical head design of pitot-static tube is recommended as being more robust and also more suitable for water-cooling, where this is necessary. The design of



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this hemispherical head pitot-static tube is shown in Fig. (5)10. Where it is possible to go into the gas stream through a bend in the duct, the pitot-static tube can be straight, otherwise the tube may be bent at a distance of at least 14 tube diameters from the pitot head.

The bent tube is, however, frequently awkward for insertion into

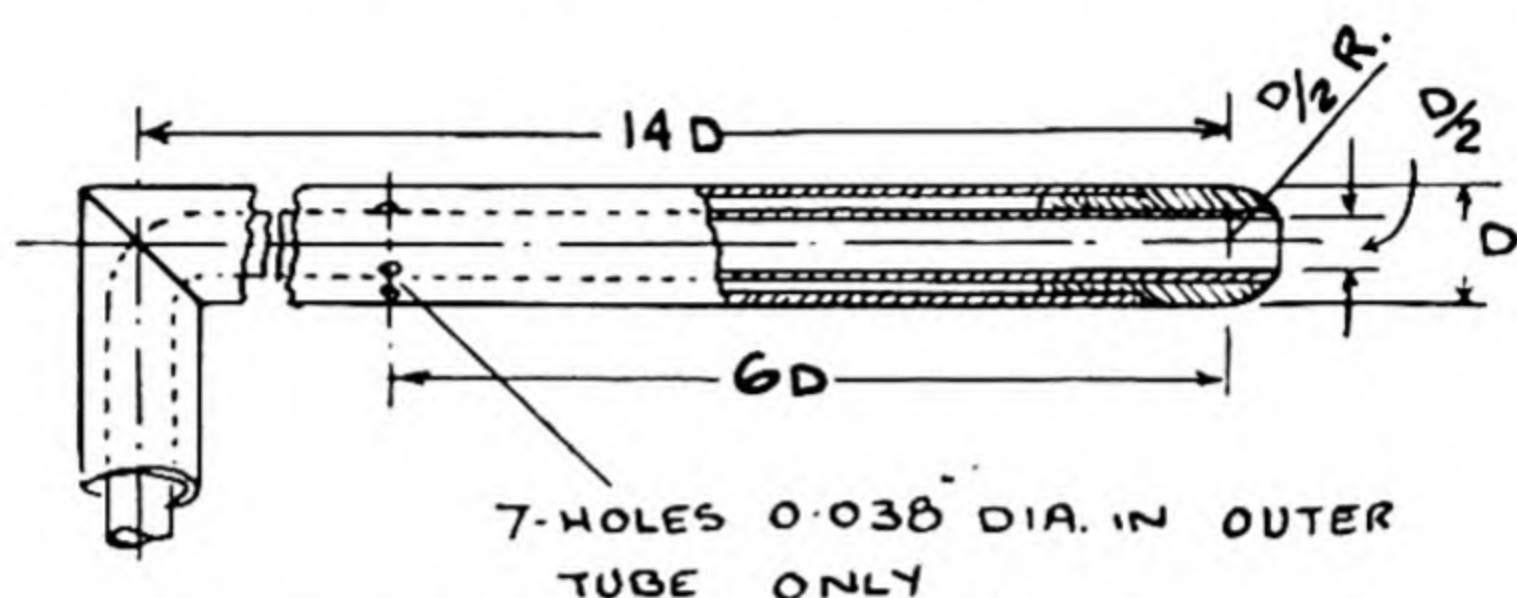


Fig. (5)10. Hemispherical Head Pitot Static Tube.

furnaces and in this case a straight tube having the total head tapping in the form of an inner tube bent round to open on one end of a diameter of the main tube and the static tube let in flush with the end, can be used, as shown in Fig. (5)11.<sup>5.13</sup> Such an instrument must be calibrated either itself or by using a model. Alternatively, the static pressure tapping can be replaced by a second tube ending on the same diameter as the total head, but at the opposite end. Because this second tapping is in the wake of the main tube, it measures a pressure which is less than the static pressure by an amount which depends on the dynamic head. Such

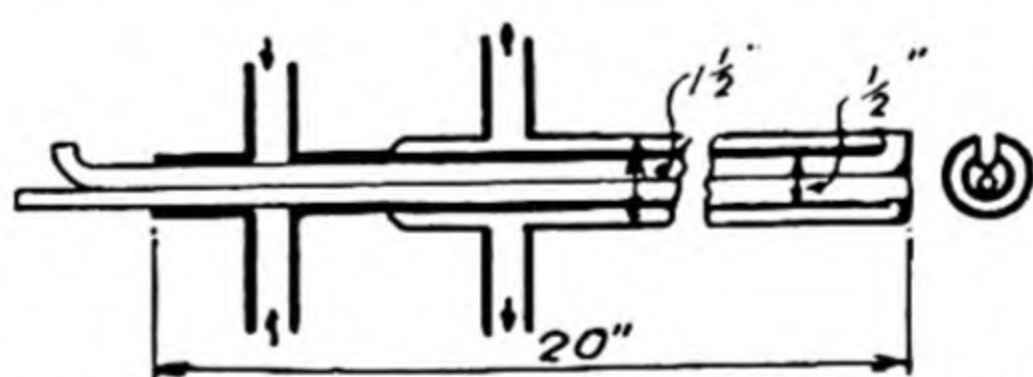


Fig. (5)11. Straight Pitot Static Tube (Kofler and Schefels).

an instrument accordingly gives a differential pressure greater than  $\frac{1}{2}\rho v^2$ .

Pitot tubes for cold air and gas can be constructed of steel or brass; uncooled steel can also be used in producer gas up to 500° C. For higher temperatures water-cooled pitot tubes are essential.<sup>5.14</sup>

5.13 KOFLER, F., and SCHEFELS, G.: "Stromungsvorlage in Herdraum Siemens-Martin Ofen," *Stahl und Eisen*, **53**, p. 162 (1933).

5.14 LECKIE, A. H.: "The Measurement of Temperature and Flow in Furnace Systems," *Fuel*, **19**, p. 42 (1940).



Besides the pitot tube, the vane anemometer and instruments which measure the force on a balanced vane can be used for velocity measurements. These instruments are not suitable for insertion into hot ducts, but can be used for measuring the air going in through an open orifice, such as the air inlet port of a furnace, or for the quantity of air flowing into a cold furnace from a port.

**5.4.5. The Dilution Method.** A method of measuring which is frequently used for liquid flow is to introduce a measured quantity of soluble material into the stream and to measure the resulting concentration at a point sufficiently far downstream for it to be perfectly mixed. In the case of water, salt may be used as the indicator material. A similar method is, in theory, applicable to gas flow measurement if a second gas can be introduced very low concentrations of which can be measured. Since the quantity of indicator gas is measured at room temperature and the ratio of indicator gas to total gas is unaffected by temperature, the dilution method of measurement necessarily measures the volume at room temperature of the gas flowing whatever may be its temperature inside the system.

The possibilities of this method for furnace measurements have been fully discussed in a paper by Rummel.<sup>5.15</sup> It is of particular value when orifices cannot be used owing to the absence of a straight stretch of channel or because the flow is pulsating. Rummel concludes, however, that the method is not easy or cheap and requires great care. The difficulty does not lie in the mixing, which can be made quite perfect by the use of a number of small jets distributed across the gas stream for the introduction of the indicating gas or by the use of obstacles which set up whirls in the gas. The main difficulty is that considerations of cost restrict the quantity of indicating gas which can be used to a very small fraction of the main gas flow and it is very difficult to measure such low concentrations with sufficient accuracy. One possible indicator is hydrogen, which is detected by means of its effect on the change of thermal conductivity of the gas. In the case of air or gases which contain no hydrogen originally, 0.25% of hydrogen is sufficient to be measured so as to give an accuracy of 2%, but even this percentage implies a great many cylinders of hydrogen to measure industrial quantities of gas. For measuring such gases as blast furnace gas, which already contains a small quantity of hydrogen,

<sup>5.15</sup> RUMMEL, K.: "Mitt. 220 der Wärmestelle," *Archiv. f. d. Eisenhüttenw.*, **9**, p. 179 (1935).



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about 2% of hydrogen must be added for accuracy. Attempts have also been made to use gases such as chlorine and  $\text{SO}_2$  as indicators ; the concentration of these gases can be measured chemically. None of these methods are, however, suitable for high temperatures or when active reaction is going on. Rummel concludes that the method is only possible for research purposes, and even then it tends to be very expensive. A modification of this method has recently been introduced which goes some way toward overcoming these difficulties.<sup>5.16</sup> In this method the indicator material is radon

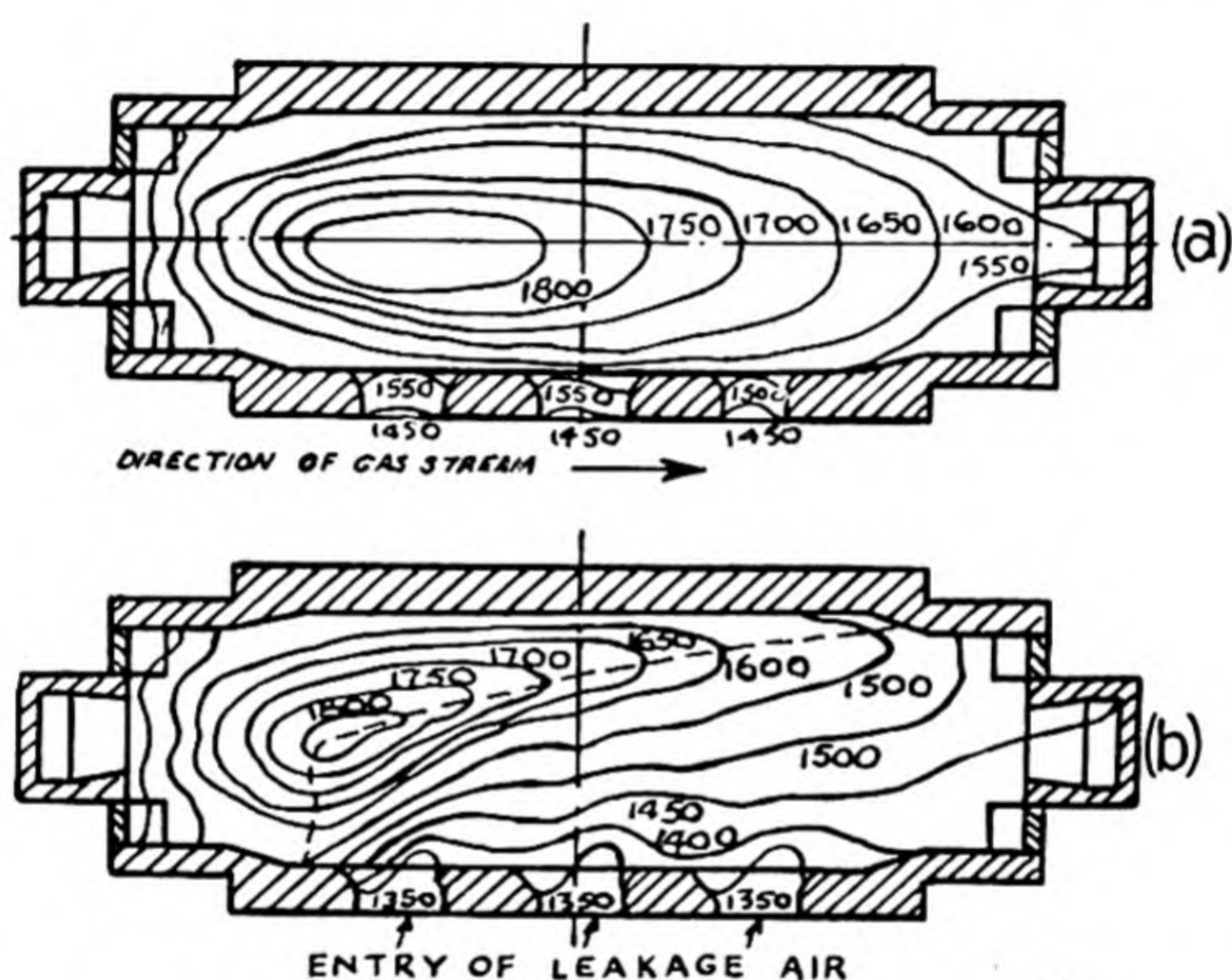


Fig. (5)12. Isotherms in an Open-Hearth Furnace: (a) without In-leakage and (b) with In-leakage. The broken line in (b) indicates the limit of leakage air.

gas, which can be detected in extremely small quantities by accurate electrical methods. For this reason radon can be used as indicator to measure large volumes of gas. In addition it is quite unaffected by temperature or chemical reactions, so that it is appropriate for all kinds of furnace measurement.

**5.4.6. Measurements of Leakage.** The quantity of air flowing into a furnace through the "skin" at points where the system is at a pressure below atmospheric can easily be a very considerable fraction of the total flow. The importance of such leakage in an

<sup>5.16</sup> MAYORCAS, R., and THRING, M. W. : "The Use of Radon to Trace the Flow Pattern of Gases in Furnaces," *Nature*, **95**, p. 66 (1943).  
MAYORCAS, R., and PERRY, K. P. : "A New Aerodynamic Technique employing Radon for Tracing Gas Flow in Hot Systems," *Proc. Conf. "Some Aspects of Fluid Flow," Institute of Physics* (1950).



actual furnace is shown by Fig. (5)12<sup>5.17</sup>; Fig. (5)12b shows the isotherms in an open-hearth furnace with a normal arrangement of doors, while Fig. (5)12a shows the corresponding field when precautions were taken to eliminate such leakage. Similarly, it was shown in <sup>5.18</sup> that the air infiltration in a producer-gas-fired furnace may be equal in volume to the producer gas flow.

The leakage into parts of the furnace system which carry fully burnt combustion gases can be deduced by measuring the increase in oxygen content of the gases at various points along the system. From the change in oxygen content along a section, the ratio of the air-infiltration in this section to the original combustion gases can be calculated and hence, knowing the quantity of the original combustion gas, the actual magnitude of the leakage can be obtained. This method cannot, however, be used for estimation of the leakage in regions where combustion is taking place, such as in the furnace chamber of the open-hearth furnace. In such a case the total leakage of air over a part of the system which includes the whole of the combustion space can still be deduced as follows. The quantity and composition of the fuel and the quantity of air entering by the normal routes are measured; the oxygen content which the combustion gases should have if there were no leaks is calculated from these measurements. The oxygen content of the gases at a point where combustion is complete is then measured and hence the total quantity of air leaking into the whole of the system between the air and fuel meters and the sampling point can be calculated. For example, in the case of the open-hearth furnace, such a method could give the total leakage through the ingoing regenerator, the uptake port and the furnace chamber, if the air and gas are measured at the valves and the oxygen content of the combustion gases at the exit port. It would not, however, be possible by this method to obtain the leakage into the furnace chamber alone unless the air and gas could be metered in the ingoing ports.

Measurement of the air infiltration through partly open doors, sight holes, etc., can also be made by the methods discussed in Section 5.4.4. Thus, for example, a pitot tube can be placed at the centres of small rectangles subdividing the area of the open doors, and the total air entering through the opening can be roughly

<sup>5.17</sup> SCHWIEDESEN, H.: "Falschluff im Ofen betrieb.," *Archiv. f. d. Eisenhüttenw.*, 9, p. 319 (1936).

<sup>5.18</sup> HALLIDAY, I. M. D.: "Preheat Temperatures of Gas and Air," *I.S.I. Special Report No. 37*, p. 48 (1946).



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calculated from the mean of these readings multiplied by the total area. This method is less accurate here than the metering by means of the pitot tube in a long pipe because in the present case the flow is much less regular.

A useful instrument for measuring the air infiltration through the brickwork of a furnace or regenerator, designed by J. E. Doyle of the United Steel Companies, is shown in Fig. (5)13. In use, the large end of the cone is placed against the brickwork and sealed with plasticine. The funnel and tube offer so small a resistance compared to that of the brick itself that they do not cause any appreciable change in the quantity of air flowing through the region

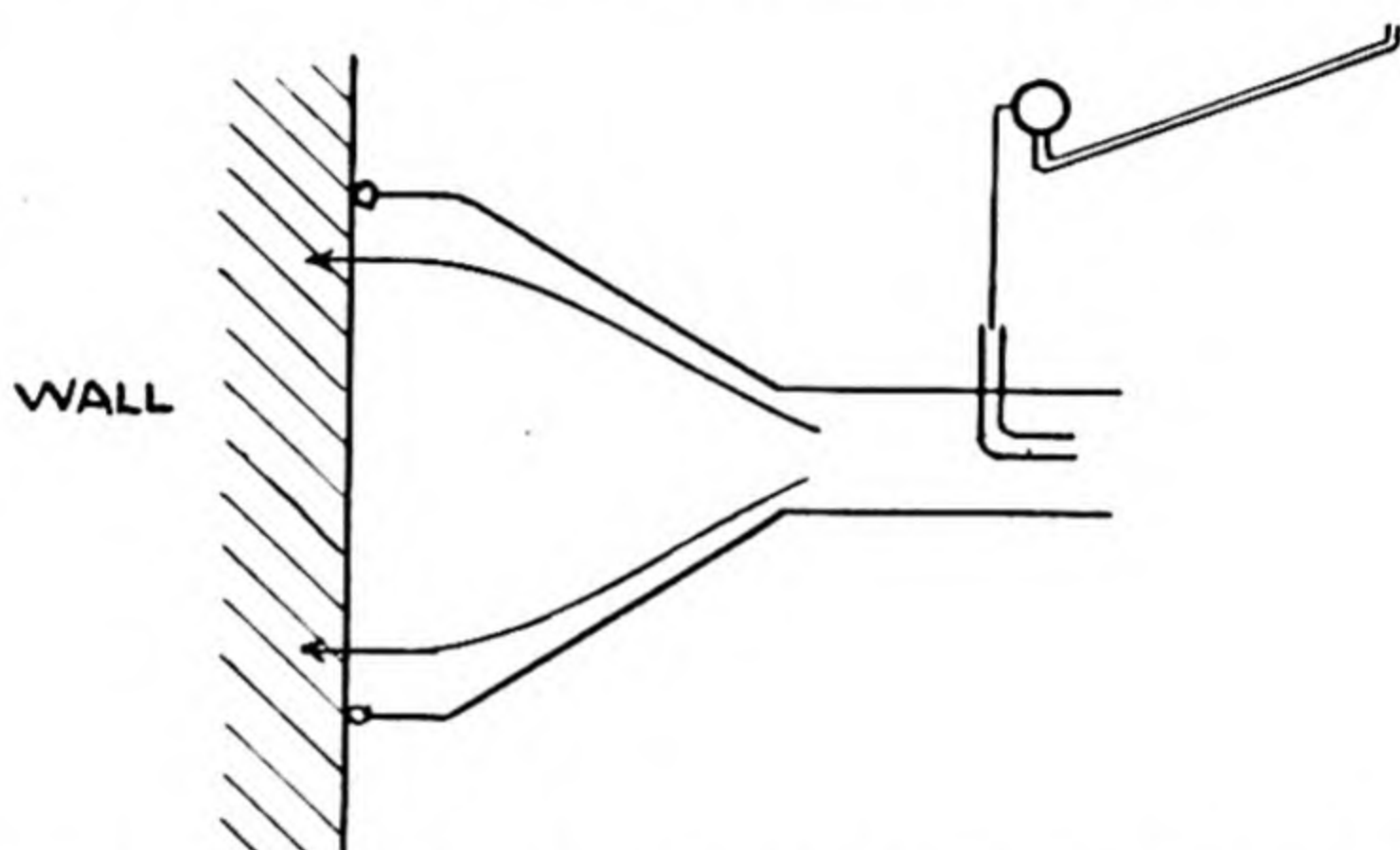


Fig. (5)13. Leakage Funnel for Measuring Air Infiltration through Brickwork.

of brick-work covered: hence they serve only to raise the velocity in the tube sufficiently for it to be measured by means of the pitot tube inserted in the entrance tube to the funnel.

#### 5.5. Total Flow in a Circuit: Representation and Summary of Results

**5.5.1. Pressure Balances for Proposed Furnaces.** In the preceding three sections (5.2, 5.3 and 5.4) all the material necessary to solve the first problem specified in 5.1.2, namely, that of constructing a balance for the total air flow, has been assembled. Where the calculations are being made for a proposed furnace such a balance has to be used to calculate the area of cross-section of various parts of the system and the height of the stack necessary to obtain the desired quantity of flow. The usual method of doing this is to make a guess at the appropriate areas and heights, calculate the resistances in terms of pressure differences for the desired flow quantity from the formulæ of 5.3, calculate the corresponding



buoyancies for the assumed temperatures and heights, and see if the aeromotive forces due to buoyancy in vertical sections where the gas is flowing upwards and due to fans or injectors are sufficient to overcome the resistances. If not, either the aeromotive force must be increased (e.g. increase in the proposed chimney height) or else the resistance must be decreased (e.g. increase in the cross-sectional areas of the parts which offer the greatest resistance). Such a calculation is conveniently set out in the form of a table similar to that given by Etherington,<sup>5.19</sup> an example of which, with actual figures, has been published by Leckie.<sup>5.20</sup> An appropriate form of table is given in Table 5.5.

TABLE 5.5

## PRESSURE LOSS CALCULATIONS FOR PROPOSED FURNACE

Part of System	Area, ft. <sup>2</sup>	Calculated velocity, ft./sec.	Estimated gas temperature, °C.	Calculated dynamic head, $\Delta p_v$ inch w.g.	Calculated friction, $\Delta p_r$ inch w.g.	Calculated buoyancy, $\Delta p_b$ inch w.g.	Net pressure loss, $\Delta p_b + \Delta p_r + p_v$ inch w.g.
1. Valve 2. Regenerator 3. Port etc.							

**5.5.2. Pressure Balances for Furnaces which are already Built : Comparison of Calculated and Measured Pressures.** In the case of furnaces which are already in operation, it is often desired to see what is happening by comparison of the calculated pressure balance with actual pressure and quantity measurements. In particular, it is of interest to see (1) whether any part of the system is badly blocked up so that its resistance is considerably greater than it should be according to the designed dimensions ; and (2) what parts of the system are the effective throttling factors which primarily determine the quantity of gas flowing through it. To do this, the furnace is divided up into parts as discussed in Section 5.5.1 and the static pressure or total pressure is measured at the junction between each of these parts relative to atmospheric pressure at the same height. A convenient way of measuring the static pressure

<sup>5.19</sup> ETHERINGTON : *Modern Furnace Technology*, p. 187 (Griffin).

<sup>5.20</sup> LECKIE, A. H. : "Factors affecting the Design of Open-hearth Furnaces," *Iron and Coal Trades Review*, Table 4 (26 Mar. 1943).



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is to bore a porous firebrick with a longitudinal hole, insert a silica tube about 2 ft. long into the brick, and place them in the hole obtained by removing a loose brick in the structure in such a way that the tube is horizontal, making sure that the inner face of the soft brick comes flush with the inner wall. The outer end of silica tube will project far enough for a rubber tube connection to the manometer to be attached without overheating. The results are then compared with the calculated pressure differences by setting them out as in Table 5.6.

TABLE 5.6

COMPARISON OF CALCULATED AND MEASURED PRESSURE LOSSES

Part of system	Calculated friction, $\Delta p_R$ inch w.g.	Calculated buoyancy, $\Delta p_B$ inch w.g.	Calculated dynamic head $\Delta p_K$ inch w.g.	Net calculated pressure loss $\Delta p_R + \Delta p_B + \Delta p_K$	Observed pressure loss
	1	2	3	4	5
Fuel-bed combustion chamber Heating chamber Recuperator, etc.					

Where the dynamic head changes appreciably across one section it is usually better to consider, for the purpose of the analysis, a longer section both ends of which are large enough for the dynamic head to be negligible, since there are likely to be errors in the measurement of the static pressure in regions where the dynamic head is large. When, however, it is necessary to take a pressure tapping in a region of high velocity it is either necessary to measure the total head by means of an inserted tube, such as that in Fig. (5)2c, or else correct the observed pressure difference for the change in dynamic head according to formula (5/8). When Table 5.6 has been constructed, columns 4 and 5 are compared. If there is a large difference between the calculated pressure loss in any part of the system and the corresponding measurement, then it follows either that the calculated friction pressure loss is greatly in error in this part or else the calculated buoyancy. In practice, however, there is not likely to be much error in the buoyancy pressure cal-



culuation as the heights are known accurately and errors in the temperature do not cause large errors in the calculation of buoyancy (*vide* Fig. (5)3). Hence the error can usually be attributed to the frictional term. If the friction losses in all the sections show a similar error, it follows that there is an error in the estimate which has been made for the total quantity flowing. For example, if all the friction losses are observed to be higher than the estimated ones, the quantity flowing is probably greater than the value assumed in the friction calculation. What is more likely to occur, however, is that most of the disagreement corresponds to a single part of the system and that this will be traced to the formation of a deposit which has appreciably reduced the area of this part of the system.

Such a table also enables the operator to see at a glance which parts of the system are primarily responsible for determining the gas flow.

**5.5.3. Graphical Pressure Balances.** When a tabular pressure balance has been constructed as discussed in Section 5.5.2, extra clarity is obtained by showing the results graphically. A convenient way of doing this has already been illustrated for a regenerative glass tank in Figs. (5) 4c and 4d. Such a figure is constructed by plotting horizontally the pressures relative to atmospheric, with a heavy vertical line for atmospheric pressure, and making successive points on the vertical axis represent the connecting points between successive parts of the system where the measurements are made. The pressures measured at all these connecting points are then plotted in and joined by straight lines. If the buoyancy pressure  $\Delta p_b$  to be expected in each part is then calculated, a line can be drawn connecting the point at the beginning of each section with the pressure which would result at the end of this section if buoyancy alone caused the pressure change. Such lines are shown hatched in Figs. (5) 14a and 14b. The difference between the abscissae of these two lines at ordinates corresponding to the ends of the various parts of the flow system represents the resistance in these parts of the system for the actual flow. Thus in the case illustrated by Fig. (5) 14a it can be seen at a glance that the only significant resistances are those which occur between the atmosphere and the base of the regenerator (air inlet valve), between the top of the regenerator and the furnace (inlet port) and between the furnace and the top of the outgoing regenerator (exit port). A similar diagram for steel furnaces has been given by Gray.<sup>5.21</sup>

<sup>5.21</sup> *I.S.I. Special Report No. 37, Section II, Pt. 3, Fig. 34 (1946).*



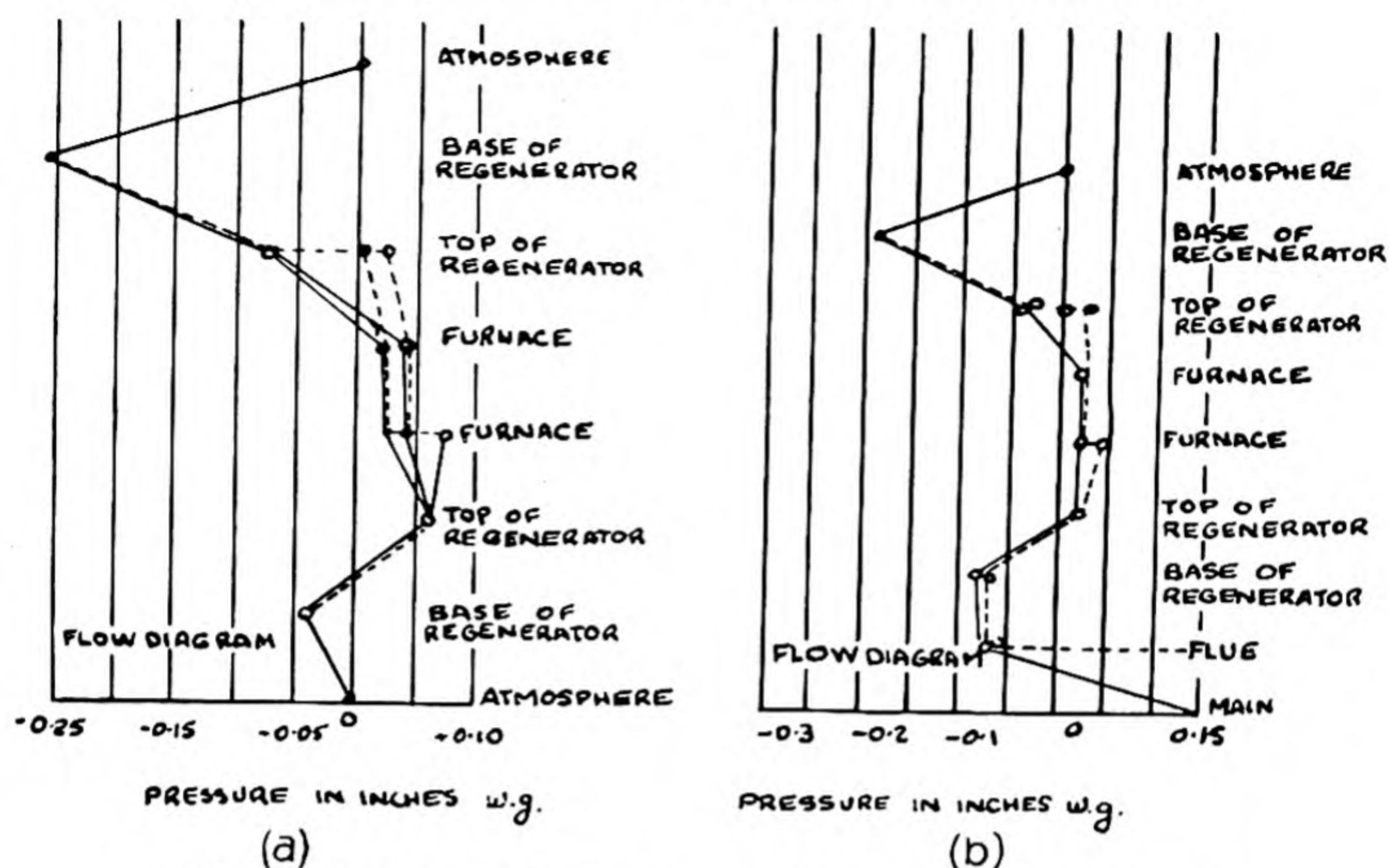


Fig. (5)14. Gas, Air and Buoyancy Pressures in a Glass Tank.

## 5.6. The Gas Flow Pattern

**5.6.1. Stratification of Flow in Horizontal Streams.** The remainder of this chapter will deal with the second of the two problems of high-temperature aerodynamics outlined in Section 5.1.2, namely, the inner structure of the gas flow: eddies, stagnant pockets, persistence of jets and subdivision between channels in parallel when this is affected by momentum (see footnote to 5.1.3).

In 5.6 a review is given of the theoretical considerations which have been applied to these problems or to similar problems in aeronautics, but it will be clear that theory has not thrown very much light on the problems so that much further work, both theoretical and on the lines of the model work and experiments on full-scale furnaces discussed in 5.7, is necessary.

The first case of importance is that of horizontal streams of gas such as occur in the furnace chamber of many furnaces, e.g. continuous and batch-type billet-heating furnaces, tunnel kilns, etc. In order to obtain the maximum heat transfer by convection to the charge, it is obviously desirable to have nearest to the batch that portion of the gas stream which has the highest velocity and the highest temperature. On the other hand, since the gases are giving up heat to the charge they tend to be cooled near it. If, therefore, the charge were situated at the top of the heating space that part of the gases which was cooled by contact with it would fall and its



place would be continuously taken by hotter gases. In practice, however, it is usually necessary to have the charge at the bottom of the heating space, and hence there is a great tendency to have immediately on top of the charge a stagnant pocket of cooler gases which reduces the convective heat transfer. This description of the process in terms of a stagnant pocket is clearly an over-simplification of the actual facts, since the real state of affairs involves a continuous variation of the velocity and temperature at points taken successively up the vertical cross-section, with lower velocities and temperatures at the lowest part which is in contact with the charge. Recent experiments with models and flow indicators in actual furnaces have shown that the so-called stagnant pocket is really a region of recirculating flow. Nevertheless, the stagnant pocket is a convenient way of describing the actual effects.

The importance of this vertical stratification which will necessarily occur in a horizontal gas stream with a heat-absorbing floor was first clearly pointed out by Groume-Grjmailo.<sup>5.21</sup> Groume-Grjmailo indicated three ways of reducing the resulting resistance to heat transfer. The first of these is to fill the whole chamber with fast-moving gas by having the chamber sufficiently small in relation to the quantity of gas. This is discussed in more detail later in the present subsection. The second method which he considered was to give a downward slope to the part of the chamber in which the gases are giving up heat, or, what amounts to the same thing, to arrange that the gases flow from a point high in the chamber down to an offtake near the floor in order to give the gas flow a vertical component. This method is a special case of the self-distributing effect which can be obtained with the downward flow of heating gases and the theory of which is discussed in Section 5.6.3. As an example Groume-Grjmailo proposed to build a tunnel kiln in the shape of an arc of a circle, highest in the centre, so that the air receiving heat as it flows in through the air-preheating zone is ascending and the combustion gases giving heat to the charge as they flow out through the charge-heating zone are descending (see Fig. (5)15). The third way of reducing the stagnant cold pocket on the base of the furnace is the introduction of the heating gas as a downwardly directed jet, which is fully discussed in Section 5.6.2.

Groume-Grjmailo investigated the possibilities of the first of his suggestions (that of avoiding the stagnant region by the use of high

<sup>5.21</sup> GROUME-GRJMAILLO: "The Flow of Gases in Furnaces," Fig. 127 (Wiley).



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velocities to fill the furnace chamber) by using models in which coloured paraffin represented the hotter gas and water the colder one. By means of those models he was able to illustrate his point clearly, and show which designs were most likely to lead to large stagnant pockets. His use of models is open to criticism, however, in two respects. (1) He does not appear to have taken precautions to adjust the Reynolds and Froude numbers for the flow in his model to values comparable to those in the actual hot furnace systems. In models in which buoyancy effects are simulated, particular attention must be paid to the Froude number, and the Reynolds number should be at least high enough to ensure turbulence, whereas his model work had streamline flow, which is why

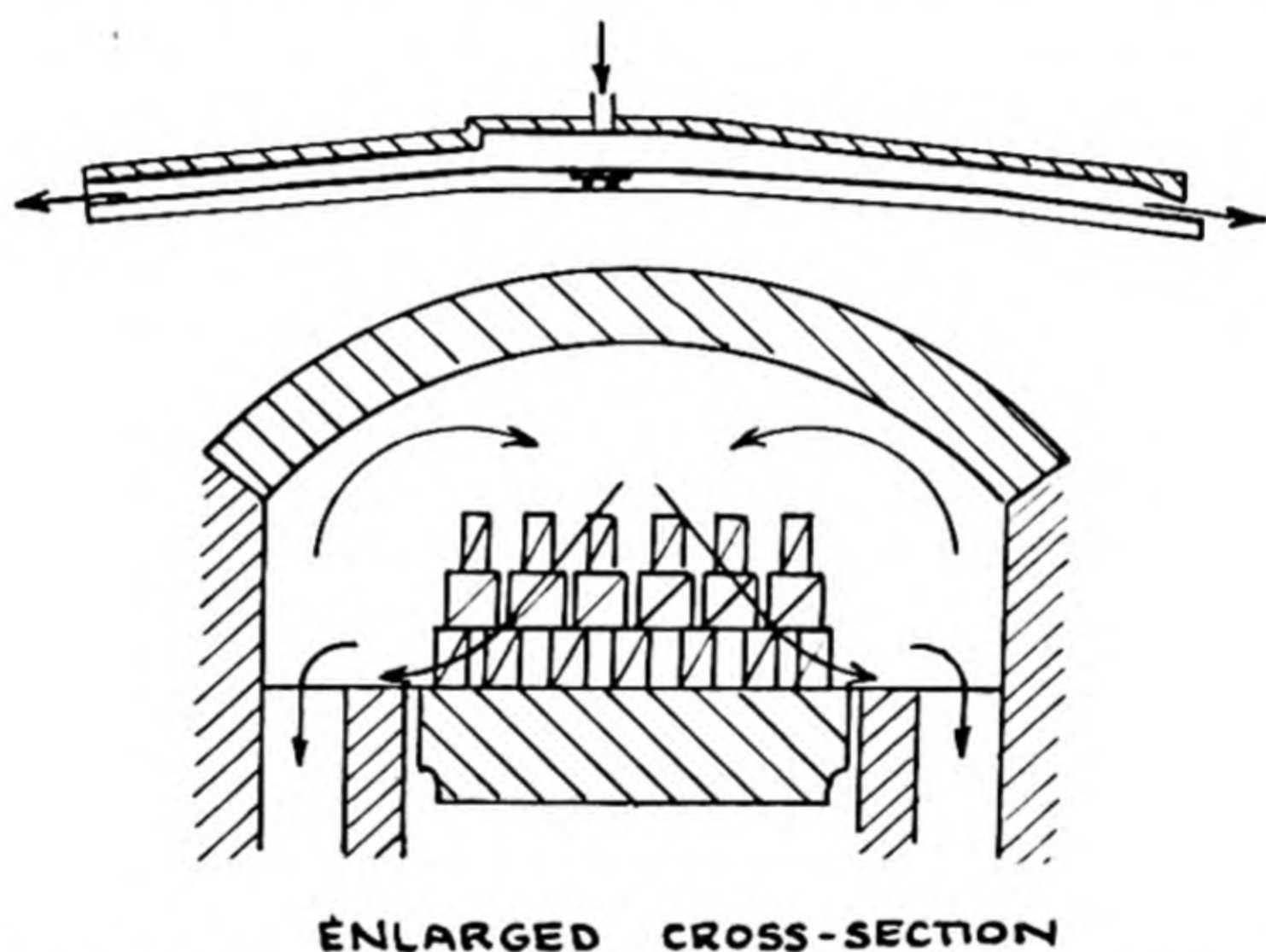


Fig. (5)15. Sections of an Arc Kiln (*Groume-Grjimailo*).

he did not observe recirculation. (2) Water and paraffin are separate immiscible fluids, whereas in an actual furnace the stagnant pocket is formed continuously from cooled combustion gases and the turbulence of the flow produces a continuous interchange of gas between it and the faster-moving stream. The general principles for the design of models are discussed in more detail in Section 5.7.4.

For numerical calculations of the height of the furnace necessary to "fill" it with gas, Groume-Grjimailo used a formula based on the assumption that the flow of hot gases in a furnace is comparable with the flow of water over a weir turned upside down, the exit port representing the weir. His calculations with this formula are, like his experiments with models, open to criticism in that they do not allow for the mixing of the hotter and colder gases; also, the



formula does not allow for the effect of the hearth on the thickness of the stream flowing in the weir. The actual formula obtained by this analogy indicated that the quantity of gas required to fill a height  $H$  of furnace is proportional to the  $3/2$  power of  $H$ . Thus, if  $H$  is expressed in feet and the quantity  $Q$  in cu.ft./sec. measured at the actual gas temperature,  $T^\circ\text{C.}$ , the formula is given as

$$H = 2.2 \times \sqrt[3]{\left(\frac{Q^2}{B^2 \cdot T}\right)} \quad \dots \quad (5/29)$$

where  $B$  is the width of the furnace in feet.

While this formula is obviously open to criticism, it has not yet been improved upon by further work and does give a reasonable qualitative relation for the effect of horizontal stratification. The subject clearly represents a rich field for further work, both on models and on the full scale, since such stratification often makes all the difference between a bad and a good furnace. It should be noted, however, that it is likely to be of greater importance where convection is the most important mechanism of heat transfer rather than radiation, since the stagnant layer is rarely opaque enough to reduce the radiation heat transfer to any appreciable extent. Hence where radiation is the predominant mode of heat transfer, elaborate precautions to reduce the stagnant layer are not very important unless the flame is so far from the charge that the angle subtended by it at the charge is reduced.

**5.6.2. The Path of a Jet.** By a jet is meant the turbulent flow of gases introduced through a small aperture into a large vessel with sufficient momentum to preserve their initial direction of flow for some distance. Jets are of importance in furnaces both as a means of overcoming the "stagnant pocket" at the base of the furnace, discussed in Section 5.6.1, and also for obtaining in a given volume a desired amount of mixing of two gases or a desired impact of air upon a bed of solid particles. The tendency of the inertia of the gas in a jet is to carry it along the axis of the nozzle; where, however, the jet gases are hotter than the surrounding gases the tendency of the bouyancy is to lift them upwards. This is a frequently occurring case where the jet is a gaseous or suspended solid or liquid combustible burning in relatively cold or relatively slow-moving surrounding air. The case is in this respect somewhat analogous to that of a water jet issuing from a hose in air, but turned upside down. In the case of a water jet or a projectile shot upwards at an



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angle  $\delta$  to the horizontal the height  $H$  to which it will rise is given by equating the loss of kinetic energy due to the loss of the vertical component of velocity to the increase of potential energy due to the ascent of the material. In the same way the equations of motion given in 5.1.3 lead to the calculation of the height  $H$  which the jet of gas will descend when it is projected downwards at an angle  $\delta$  with the horizontal, provided all effects of friction and entrainment of the surrounding gases can be neglected.  $H$  is calculated by equating the loss of energy due to the loss of the vertical component of velocity to the increase of potential energy due to the descent of the hotter gas. This latter clearly depends on the difference of the density of the jet and surroundings. Groume-Grjmailo showed that such energy considerations result in the formula

$$H = \frac{U_o^2 \sin 2\delta}{2g} \times \frac{273 + T_s}{T_f - T_s} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (5/30)$$

where  $T_s$  is the temperature of the surrounding gases ; and  $T_f$  the temperature of the jet.

Calculations of the path of the gases in an open-hearth furnace at various temperatures, based on it, are shown in Fig. (5)16.<sup>5.13</sup>

In the case of jets, however, neglect of the mixing of the jet with the surrounding gas leads to large errors, as has been shown in a very comprehensive way by R. F. Davis.<sup>5.23</sup> Unlike a jet of liquid, a gas jet has no precise boundary, but is surrounded by a turbulent region in which there is a uniform gradient of velocity down to that of the surrounding gas. A gaseous jet draws gas in from the surroundings and gives it forward velocity, consequently, by the law of conservation of momentum, the axial velocity of the jet decreases rapidly with distance from the nozzle quite apart from any change of velocity due to buoyancy effects. Thus the jet issuing from a nozzle spreads outwards in a roughly conical form with an indefinite boundary through which it continuously draws more and more gas, losing kinetic energy by turbulence as it spreads. Davis himself and other workers, e.g. Cleve,<sup>5.24</sup> have shown that if the boundary of the jet is defined as the region of any particular velocity this boundary diverges in a cone having an included angle lying between  $7\frac{1}{2}^\circ$  and  $10\frac{1}{2}^\circ$ , and this angle is practically independent of the jet velocity provided the latter is sufficient to make the jet

<sup>5.23</sup> DAVIS, R. F. : "The Mechanics of Flame and Air Jets," *Proc. Inst. Mech. Eng.*, 137, p. 11 (1937).

<sup>5.24</sup> CLEVE : "The Operation of Turbulent Air Jets," *C.A.M.A. Trans.*, No. 2.



turbulent, a condition which is always satisfied in furnace jets. Davis shows that in such a jet, the axial velocity

$$V_a = \frac{8.4dv_o}{x+2d} \quad \cdot \cdot \cdot \cdot \cdot \cdot (5/31)$$

where  $x$  is the length along the jet ;

$d$  is the initial diameter of the jet expressed in the same units as  $x$  ; and

$v_o$  is the average velocity of the jet as it leaves the nozzle.

This formula applies for all distances  $x > 8d$ . (See also Section 3.4.5 for a more detailed treatment of the velocity distribution in jets.)

If, now, such a jet is introduced into a chamber the path will be the resultant of this decreasing jet velocity and of the buoyancy or

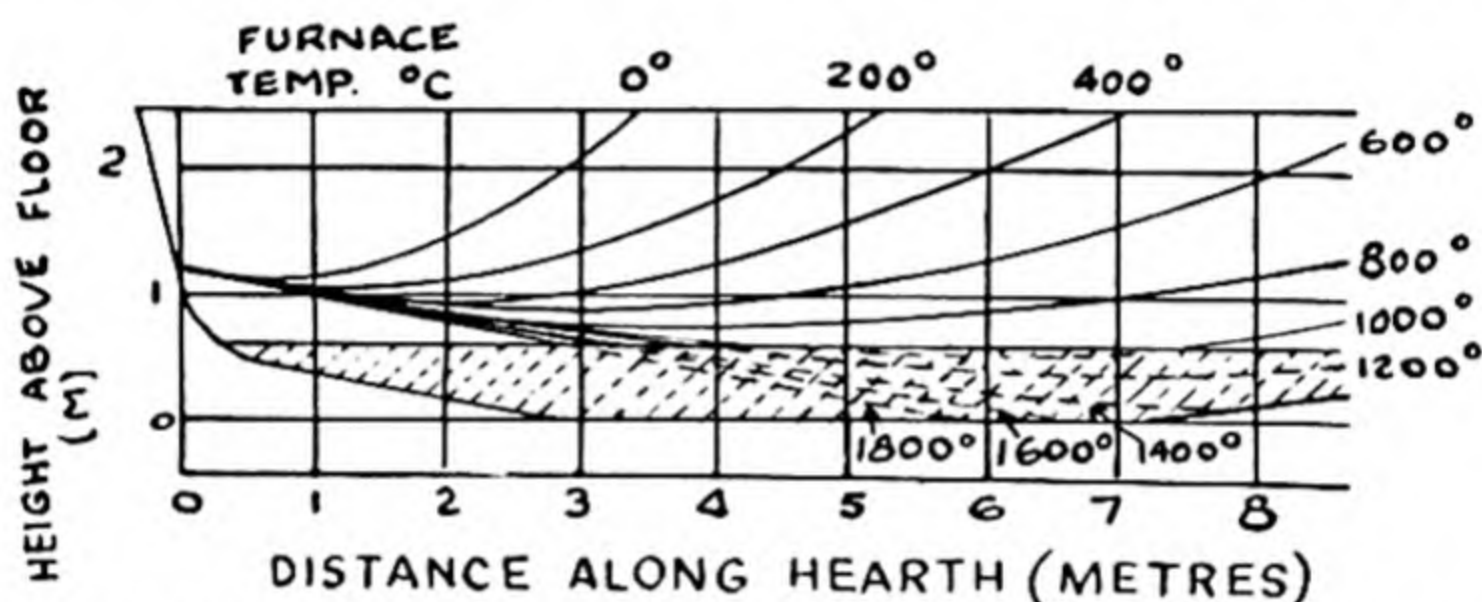


Fig. (5)16. Groume-Grjimaio's calculated paths for flame jet in an Open-Hearth Furnace when the furnace is at various temperatures.

any superimposed flow in the chamber. Davis considers first, the case of a jet entering a chamber at an angle  $\phi$  with the  $x$ -axis. The gases in the chamber have a superimposed flow of velocity  $v$  along the  $y$ -axis. In this case, the shape of the trajectory is given by

$$y = A \cdot v + x \tan \phi$$

where

$$A = \frac{x(x+4d \cos \phi)}{17ad v_o \cos^2 \phi} \times \left( \frac{T_o}{T_s} \right)^{1/3} \quad \cdot \cdot \cdot \cdot \cdot (5/32)$$

$T_o$  is the initial temperature of the jet ( $^{\circ}$  K.) ; and

$T_s$  its temperature at the point  $(x, y)$  ( $^{\circ}$  K.).

He obtains the effect of the change in temperature by assuming that on heating up the jet expands uniformly in all directions so that its longitudinal velocity is increased in the ratio of the cube root of the volume increase. Davis obtains reasonable agreement between this formula and the shape of the flame in the space above the chain-grate stoker as measured with various percentages of over-fire air projected through a jet. In the case of a jet entering a



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stagnant chamber where it is acted on by a buoyancy force, given by  $\frac{\rho_s - \rho_f}{\rho_f} g$ , Davis obtains the formula

$$y = A \left( \frac{A}{2} \cdot \frac{\rho_s - \rho_f}{\rho_f} \cdot g + v \right) + x \tan \phi \quad . \quad . \quad (5/33)$$

where  $A$  has the same meaning as in equation 5/32 and  $\rho_s$  and  $\rho_f$  are the densities of the gases in the jet and furnace respectively at N.T.P.

In this case also he obtained reasonable agreement with the

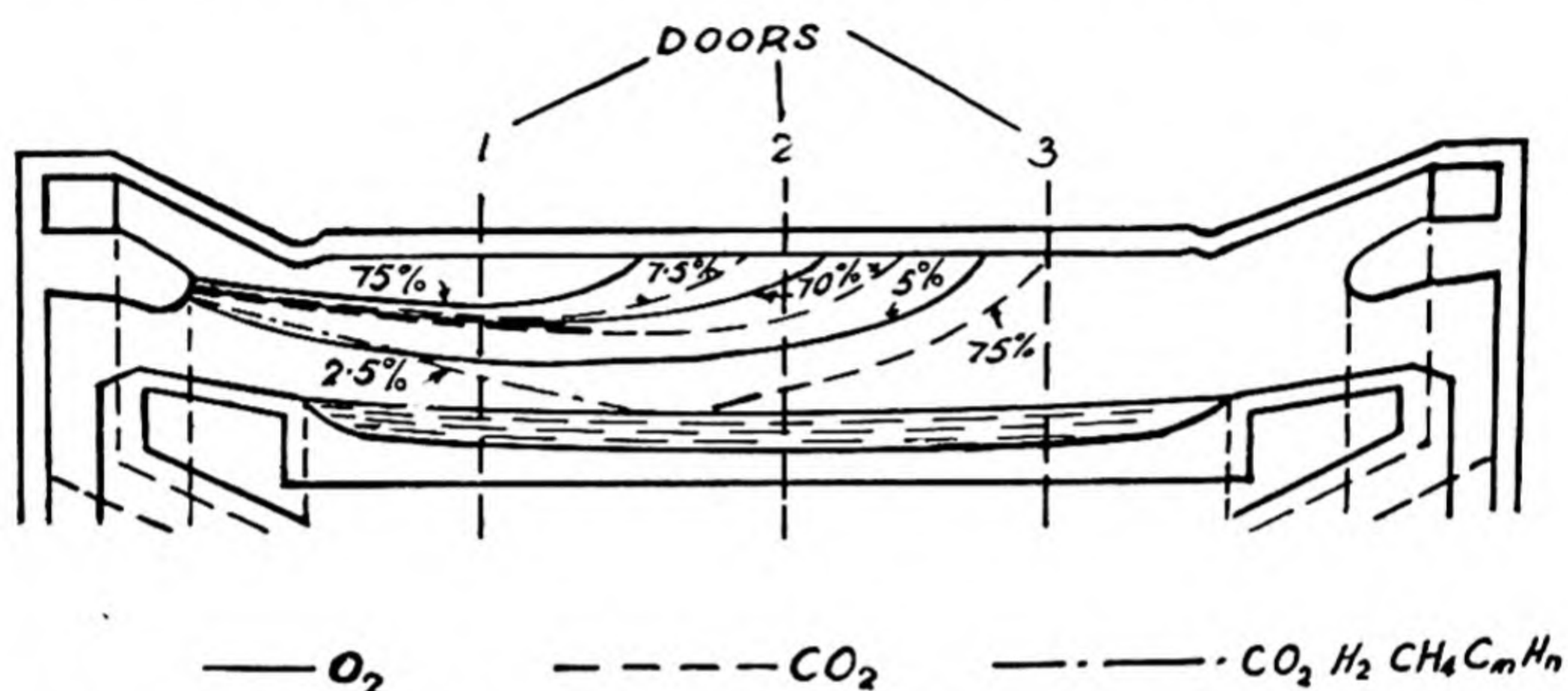


Fig. (5)17. Distribution of Oxygen, Carbon Dioxide, and Carbon Monoxide in an Open-Hearth Furnace.

distance that a pulverised coal flame descends before turning upwards from a fantail burner. Measurements of the behaviour of jets in open-hearth furnaces are shown in Fig. (5)17.\* In the paper referred to the flame paths, without and with water-cooled ports respectively, are shown, the greater persistence of the jets due to higher velocity in the latter being evident.

Chesters<sup>5.25</sup> has recently studied the recirculation pattern surrounding a jet when the latter issues into a region with walls fairly close to it. This recirculation arises when the entrainment of the surrounding atmosphere exceeds the replacement of this atmosphere by any movement which it may have in the same direction as the jet, and the principle of momentum conservation could accordingly be applied to its calculation. Chesters has produced beautiful flow patterns showing such recirculation for two-dimensional flow with various simple geometrical arrangements of the walls; Fig.

\* Figs. 5(16) and (17) are taken from <sup>5.13</sup>: *Stahl und Eisen*, 53, Pt. 1, p. 162, Figs. 6 and 7 (1933).

<sup>5.25</sup> CHESTERS, J. H., HOWES, R. S., HALLIDAY, I. M. D., and PHILIP, A. R.: "Flow Patterns in Open-hearth Furnace," *J.I.S.I.*, p. 385 (Aug. 1949).



(5)18 shows his results for a triangle with flow in through one vertex and out through the other two vertices.

**5.6.3. The Effect of Buoyancy on the Distribution of Gas-flow when the Flow Direction has a Vertical Component.** In furnaces where the gas flow is horizontal the main difficulty is, as discussed in Section 5.6.1, to ensure that the gas velocity near the floor is

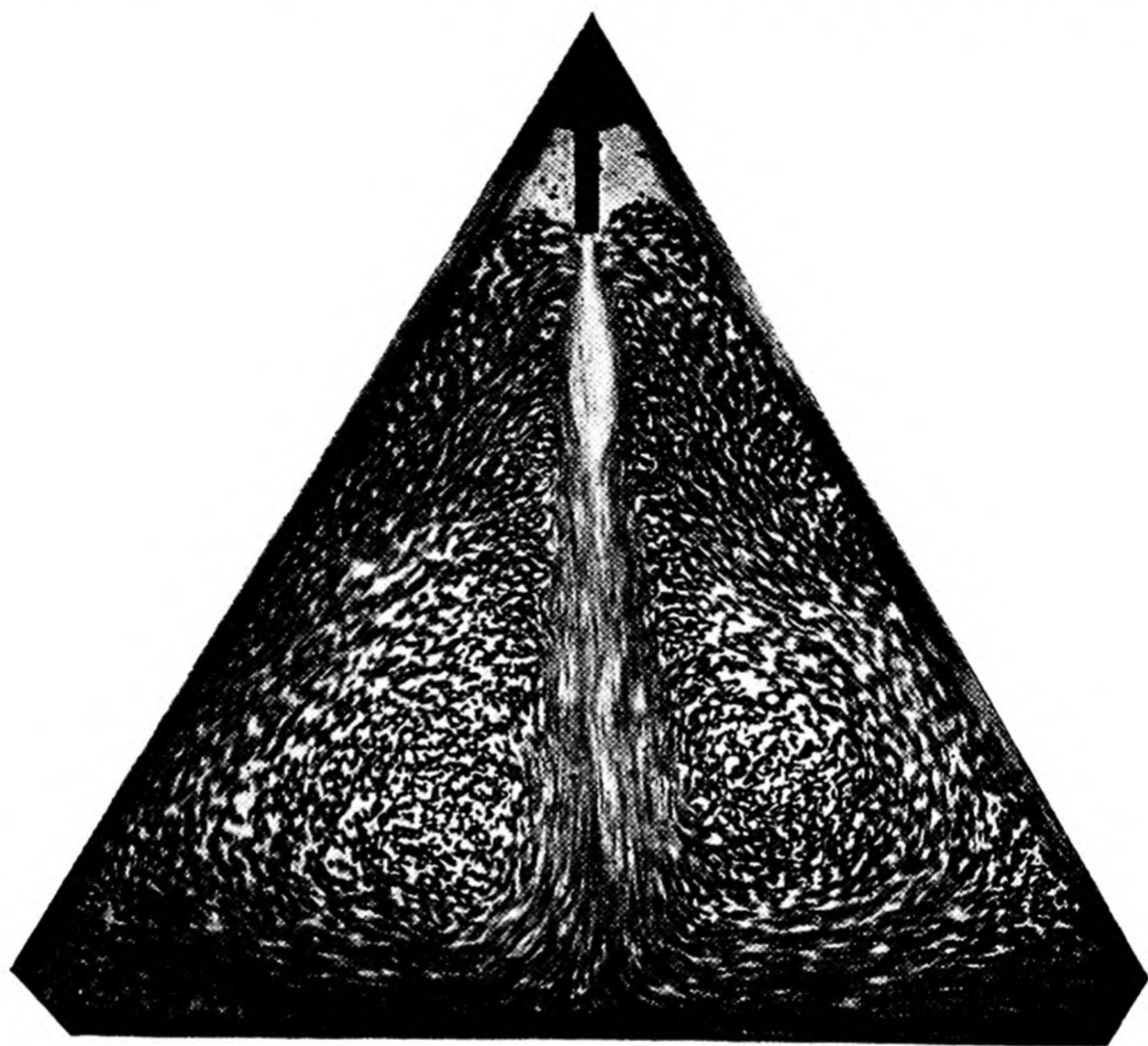


Fig. (5)18. Pattern flow in triangular space from a jet at one corner, leaving at the other two corners.

*(Courtesy of Dr. J. H. Chesters, United Steel Companies Limited.)*

high, and buoyancy always works against this. In furnaces where the gas flow is vertical the aim is usually to obtain a uniform distribution of flow right across the horizontal cross-section. This applies, for example, to regenerators and to the beehive brick kiln where it is desired to obtain uniform heat transfer between gases and bricks at all points.

One way of achieving this is to ensure that the packing is



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equivalent to a number of equal channels all having a relatively high resistance, so that the flow distribution is governed by this resistance. Then provided the differences in resistances of the various channels are small in comparison with the total resistance of a channel a uniform flow will result. This method is not possible, however, where the available aeromotive force is inadequate.

Groume-Grjmailo pointed out that in such a case buoyancy can either help or hinder the designer in achieving his aim, according to whether the flow of gases is up or down. He showed that if a current of gas which is hotter than the surrounding walls is flowing downwards it will automatically tend to distribute itself uniformly in a single wide channel or to subdivide nearly equally between channels connected in parallel. To prove this, consider the case of two channels *A* and *B* and suppose for example that the inertia of the gas in the common flue at the top tends to carry it down *B* more than *A*. Then *B* will become hotter than *A* and the gas will be less cooled as it flows down. Hence the buoyancy pressure  $\Delta p_B$  which opposes the downward flow will be greater in *B* and this will favour flow in *A*. Conversely, if the gas which is hotter than its surroundings is flowing upwards any tendency to uneven flow will be accentuated by the buoyancy. In the case of a gas which is colder than its surroundings buoyancy will assist even distribution when the flow is upward, and hinder it when the flow is downward.

This principle is of great importance in furnace design and since the arguments are perfectly general will apply whenever buoyancy in a part of the system is an important factor in governing the flow distribution. Clearly, the regenerators in an open-hearth or glass tank satisfy it since the heated gases flow upward and the heating gases downward. Similarly, in a beehive kiln the gases flow to the top around the outside of the charge, and then, in the part where even heating is desirable (their passage through the charge) they flow downwards to a series of offtakes in the floor. Likewise in the tunnel kiln Groume-Grjmailo considers that if the brick heating region is sloped so that the gases are flowing partly downward, this principle will come into operation and it will be easier to make the gases pass through the parts of the charge which are in contact with the cars. It is a little doubtful, however, whether an appreciable effect would be obtained unless the angle of slope were made fairly steep. Some soaking pits make use of this principle, others have cross firing and rely on the fact that the ingots stand up nearly to the top of the furnace.



Finally, it is possible to obtain uniform distribution in vertical flow by special adjustment of the resistances of different parts so that they compensate for the tendency of momentum to cause maldistribution. In this way good results can be achieved without the use of very high resistances, but no general rules can be given. Each case must be adjusted in operation or designed specially by experiments on a model.

**5.6.4. The Speed of Mixing of Two Streams.** It has been shown in Section 3.3.3 that the speed of combustion of diffusion flames in furnaces depends entirely on the rate of mixing of the combustion gas and air, and the work of Rummel on the mixing and combustion processes was discussed there. In this sub-section the mechanism of the mixing process of two streams flowing side by side is discussed in more detail.

Just as thermodynamics can give the equilibrium of chemical reactions but not the speeds, so the laws of dynamics discussed in Section 5.1.3 give the ultimate flow velocities of mixed streams but cannot give the rate of mixing. For this the detailed mixing mechanism must be known.

In purely laminar flow the only mixing process is that of true diffusion. According to the kinetic theory of gases, diffusion is the mixing of gases by the actual free movement of molecules between collisions, such movement carrying molecules of the one gas into the region mainly occupied by the other by a purely random process. Each molecule is assumed to travel between collisions an average distance known as the mean free path. Reasonably good agreement between the measured diffusion coefficients and the calculated values based on the mean free path of the gases is obtained, in just the same way as a reasonable account of gaseous viscosity can be obtained on the basis that the different flow velocities are carried across the boundary by moving molecules. The exact equation for the rate of mixing by diffusion can be solved provided the diffusion coefficients concerned are known. The diffusion equation is that the number  $N$  of molecules carried across unit area perpendicular to the axis  $y$  in unit time is given by :

$$N = D \frac{\partial n}{\partial y}$$

where  $n$  is the concentration of the molecule concerned at any point and  $D$  is the diffusion coefficient.



In the case of turbulent flow, mixing which occurs in furnaces is, however, as shown by Rummel, some thousands of times greater than when molecular diffusion is the only factor. Aerodynamic theory shows, however,<sup>5.26</sup> that turbulent flow may be compared with ordinary laminar flow if the mixing and transfer of energy or momentum is supposed to be carried out not by single molecules but by small eddies or pockets of gas which carry the property concerned for an average distance,  $L'$ , known as the mean mixing length (comparable to the mean free path of a gas molecule in laminar flow). It may be shown that the mean rate of transfer of any property  $\phi$  carried by the eddies across a plane perpendicular to the  $y$  axis is equal to

$$L'v \cdot \frac{\partial \phi}{\partial y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5/34)$$

This equation is analogous to the ordinary diffusion equation with the coefficient of diffusion replaced by  $L'v$ ;  $v$  is the root mean square of the turbulent component of motion in the  $y$  direction. The transfer of momentum according to this equation results in a resistance to the setting up of velocity gradients which is much greater than that caused by ordinary viscosity. Similarly, the transfer of heat by eddies is much greater than would occur by normal molecular conduction.

The mean mixing length is a measure of turbulence, hence when it can be evaluated the mixing process can be calculated. Conversely it is possible to calculate the mixing length from measurements of the mixing rate. Although this subject has not yet been fully worked out it does show the close interaction of aerodynamic theory with furnace practice. A method of measuring the mixing length in actual furnaces is discussed below in Section 5.7.3.

In the theory of airfoils, two suggestions have been put forward to enable the mixing length to be calculated. One is that in a given flow the mixing length is proportional to the distance of the point concerned from the wall, and the other, that it depends on the distribution of mean velocity. It would be of considerable interest to apply these assumptions to Rummel's results and other mixing systems of interest in furnaces. In the next section they are applied to the case of entrainment of surrounding gas by a jet.

Schiegler<sup>5.27</sup> has studied experimentally the case where two

<sup>5.26</sup> GOLDSTEIN: *Modern Developments in Fluid Dynamics*, p. 205.

<sup>5.27</sup> SCHIEGLER, L.: "Der Strömungsvorgang in der Brennkammer von Rostfeuerungen," *Z.V.D.I.*, **82**, p. 849, and **83**, p. 995.



parallel streams of the same gas enter a chamber side by side with different velocities. He found that each stream retains its velocity for a considerable distance on the side remote from the other stream, but that they have a region of width  $y$  between them in which there is a continuous transition from the one velocity to the other.  $y$  increases in proportion to  $h$ , the length of the path from the point of introduction. Over a certain range of Reynolds numbers (7000–60,000) his results can be expressed by the following approximate formula

$$\frac{y}{h} = 0.3 - 0.03 \times \frac{v_1}{v_2} \quad . \quad . \quad . \quad . \quad . \quad (5/35)$$

where  $v_1$  and  $v_2$  are the lower and higher velocities respectively. He has applied this result with some success to the mixing of the gas stream over a fuel bed, where different velocities arise due to the compartmenting of the air stream below the bed.

Recent work on models of open-hearth furnaces<sup>5.28</sup> has shown that the mixing of gas and air streams is greatly dependent upon the design of the furnace ports and their relative positions.

When the flow is not only turbulent but also “eddying,” as defined in Section 5.1.5, the mixing becomes still more rapid. This explains the rapid combustion rate which can be caused by obstacles in diffusion flames.

**5.6.5. The Speed of Entrainment of the Surrounding Fluid by a Jet.** There is one case where the geometrical simplicity of the system enables a set of formulæ for the velocity field and the mass of entrained fluid to be developed from laboratory experiments. They are both useful to furnace designers for calculating the combustion rates (in so far as these are governed by mixing rates of fuel and air, see 3.3.3 and 3.5.4) and to theoretical physicists for comparing different theories of turbulent mixing. This is the case of the free jet issuing into an infinite stagnant surrounding fluid and entraining this fluid as it flows so that its velocity continually falls off but the mass flowing continually increases while the momentum is constant along the jet because there is assumed to be no longitudinal pressure drop. Various shapes of jet have been studied theoretically and experimentally but the case of most interest here is that of the cylindrical jet.

Certain facts can be concluded about this type of jet from

<sup>5.28</sup> COLLINS, R. D., and TYLER, J. D.: “Gaseous Mixing in Open-hearth Furnace Models,” *J.I.S.I.*, **162**, Pt. 4, p. 457 (Aug. 1949).



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symmetry and the fundamental laws of hydrodynamics before any assumptions are introduced as to the mixing process. Let  $u$  be the velocity parallel to the main flow of the jet at any distance  $x$  from the jet mouth measured along the jet axis and at a distance  $y$  from the jet axis. The diameter  $d$  of the jet pipe mouth is assumed first to be very small compared to  $x$ . Then from the conservation of momentum  $G$  is a constant

$$G = \rho \int_0^{\infty} 2\pi u^2 y dy \quad . \quad . \quad . \quad . \quad (5/36)$$

If the jet is assumed to be of conical shape the velocities at any two points on a line through the jet mouth must have the same ratio to the velocities at the same values of  $x$  on the axis; hence  $u$  must be of the form

$$u = f(x) \cdot \phi(y/x)$$

By writing  $\eta = y/x$ , (5/36) becomes

$$G = [f(x)]^2 \rho \int_0^{\infty} 2\pi [\phi(\eta)]^2 \eta x^2 \cdot d\eta$$

and the right hand side can only be a constant for all  $x$  if

$$f(x) = 1/x$$

so  $u = 1/x \cdot \phi(y/x) \quad . \quad . \quad . \quad . \quad (5/37)$

It follows at once that on the axis where  $y=0$  or at points on a line through the jet orifice for which  $y/x = \text{constant}$ ,  $u$  varies inversely as  $x$  the axial distance of the given point from the orifice.\*

The point which has been of great interest to the theoretical physicists is the form of the function  $\phi(y/x)$ . This has been determined experimentally by many workers<sup>5.29</sup> and has been calculated by :

\* This fact is illustrated by the formula (5/31) already quoted for the axial velocity of a jet obtained by Davis, assuming that a jet issuing from an orifice of diameter  $d$  is equivalent to one issuing from a point a distance  $2d$  further back.

5.29 TRÜPEL, TH.: Über die Einwirkung eines Luftstrahles auf die umgebende Luft, *Z. f. d. ges. Turbinenwesen*, **12**, 52, 66 (1915).

BETZ, A.: *Ergebnisse der Aerodynamischen Versuchsanstalt zu Göttingen*, **11**, 69 (1923).

RUDEN, P.: Turbulente Ausbreitungsvorgänge im Freistrahle, *Naturwiss*, **21**, 375 (1933).

REICHARDT, H.: Gesetzmässigkeiten der freien Turbulenz, *V.D.I. Forschungsheft*, No. 414 (1942).

CORRSIN, S.: Investigation of Flow in an Axially Symmetrical heated Jet of Air. *Nat. Adv. Comm. f. Aeron., Adv. Conf. Rep.* 3L29 (1943), *Wartime Report W-94*.



- (i) Tollmien<sup>5.30</sup> using the Prandtl hypothesis that the shearing forces in the jet are resisted by the transport of momentum by eddies which move a distance  $L$  (the mixing length) before losing their identity. He assumes that  $L$  is proportional to  $x$  ( $L=cx$ ), giving an effective turbulent viscosity

$$\nu' = c^2 x^2 \left( \frac{\partial u}{\partial y} \right) \quad . \quad . \quad . \quad . \quad . \quad (5/38)$$

From these assumptions it is possible to calculate  $\phi(y/x)$  with  $c$  as the only unknown parameter. By fitting the resulting curve for  $u/u_0$  where  $u_0$  is the velocity on the axis at the same value of  $x$  at one value of  $y/x$  the theory can be tested by the fit elsewhere and the value of  $c$  evaluated. Tollmien found  $c=0.0158$  and the corresponding radius of the jet (the boundary outside which the undisturbed fluid has no forward velocity) was  $y_0=0.214x$  giving a jet half-angle of  $\tan^{-1} 0.214$  or approximately  $12^\circ$ .

- (ii) Howarth<sup>5.31</sup> applied the Taylor hypothesis that it is vorticity which is transported a distance  $L$  unchanged and provides the resistance to shear. This theory gives a function for  $u/u_0$  which fits the experimental results slightly worse than the Tollmien function, particularly at the edges of the jet.
- (iii) In one of the most recent papers on the subject Hinze and van der Hegge Zijnen<sup>5.32</sup> give further detailed experimental measurements and show that as far as the velocity distribution is concerned the most accurate prediction is provided not by the Prandtl or the Taylor hypothesis but by the original concept of Boussinesq. Boussinesq postulated that there is an effective turbulent kinematic viscosity (coefficient of shearing stress)  $\nu'$  which is constant for all points in the jet and Hinze found that this postulate fits the experimental results well if he puts

$$\nu' = x \cdot u_0 \, 0.00196 \text{ ft.}^2/\text{sec.} (\text{cm}^2/\text{sec.}) \quad . \quad . \quad . \quad (5/39)$$

(since  $x \cdot u_0$  is constant along the jet by (5/37)).

5.30 TOLLMIE, W.: Berechnung Turbulenter Ausbreitungsvorgänge. *Z. f. angew. Math. u. Mech.*, **6**, 468 (1926).

5.31 HOWARTH, L.: Concerning the Velocity and Temperature Distributions in Plane and Axially Symmetrical Jets. *Proc. Cambridge Phil. Soc.*, **34**, 185 (1938).

5.32 HINZE, J. O., and VAN DER HEGGE ZIJNEN, B. G.: Transfer of Heat and Matter in the Turbulent Mixing Zone of an Axially Symmetrical Jet. *Appl. Sci. Res.*, Vol. A1, p. 435.



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It is therefore recommended that for calculations of mixing rates of injector type flames where the conditions are approximately similar to those of a free jet in an infinite stagnant surrounding medium Hinze's results be applied. He obtains the velocity profile given by

$$\frac{u}{u_o} = \frac{1}{(1 + 64y^2/x^2)^2} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (5/40)$$

where 
$$u_o = \frac{6.39v_o d^*}{x} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (5/41)$$

$d$  being the jet-pipe diameter,  
 $v_o$  the initial mean velocity in the jet pipe mouth, and  
 $x$  is measured from a distance  $0.6d$  back from the jet mouth.

From (5/40) it follows by integration across the jet that

$$G = 0.0164 \rho u_o^2 x^2 \text{ ft.-lb./sec.}^2 (\text{cm.-gm./sec.}^2) \quad \cdot \quad (5/42)$$

while the mass of jet and entrained air flowing at any distance is given by

$$M = 0.049 \rho u_o x^2 \text{ lb./sec. (gm./sec.)} \quad \cdot \quad \cdot \quad (5/43)$$

If  $u_o$  is eliminated from these equations, then

$$M = 0.38x \cdot \sqrt{G \cdot \rho} \text{ lb./sec. (gm./sec.)} \quad \cdot \quad \cdot \quad (5/44)$$

While the numerical constant in this equation is likely to be appreciably in error especially when temperature variations and density differences between the jet and entrained air exist, the fact that the value of  $M$  (the total mass of the jet) is linearly proportional to  $x$  (the distance from the origin of the jet) is likely to be approximately true in all cases. This means that the amount of combustion air entrained by the fuel jet increases linearly along the jet so long as the combustion air available remains in excess of that calculated as entrained from equation (5/44).

In applying formula (5/44) it is necessary to remember that this entrained air is not fully mixed with the jet at the first moment of entrainment as the jet consists of a central portion containing a high proportion of fuel and an outer portion with a high proportion of air.

**5.6.6. Flow Distribution across Beds of Broken Solids.** In the discussion of the resistance of beds of broken solids to gas flow

\* A comparison of Hinze's formula with that of Davis (5/31) shows that the coefficients are only reliable to the order of 10%.



(5.3.4) it has been shown that the overall flow conditions are very strongly influenced by the presence of channels and irregularities of packing. The distribution of flow is still more markedly influenced by these factors: a common example is the tendency of gases to flow up the walls of a gas producer. This subject has only recently been investigated and no final conclusions have been reached. Two papers are, however, of sufficient interest to be worth mentioning.

An investigation was undertaken jointly by the B.I.S.F. and the B.C.U.R.A. in 1938 into the flow resistance and flow distribution of cold air in a commercial gas producer filled with coal, and in a dynamic water model in which broken glass was used to take the place of the fuel. It was found that with a regular packing the air spread out hemispherically from the blast hood until the whole cross-section was filled and then flowed uniformly upwards. In other words uniform packing acts as an almost perfect flow distributor. A change in the nozzle itself, on the other hand, affected pressure drop greatly but hardly altered the flow distribution. If the blast rate were sufficient to lift the fuel in one part of the bed (a condition which is naturally achieved at a much lower Reynolds number in a water model than where the fluid is air), then the uniformity of flow was completely upset.

It has since been shown by Bennett and Brown<sup>5.33</sup> that in fuel beds in active combustion the effect of the consumption of the fuel particles is to change the packing and create channels through the bed. This effect is self-aggravating, since the existence of a small channel will cause increased combustion in itself and hence increase of the channel. Such a channel will therefore persist until the fuel surface is actively disturbed or until gravity causes a collapse of the bed structure. In this paper it is also shown that the arrangements of the air inlets are very critical in regard to the resistance and flow distribution in the bed and that the direction of the gas flow in relation to that of gravity has an appreciable effect. Bennett and Brown describe a technique for studying gas-flow distribution using air with a small quantity of iodine and coating the glass particles representing the packing with starch. The starch is discoloured by the iodine in successive layers and photographs of the model after various exposure times give a valuable picture of the isochrome lines.

<sup>5.33</sup> BENNETT, J. G., and BROWN, R. L.: "Gas Flow in Fuel Beds," *J. Inst. F.*, 13, p. 232 (1939-40).



## 5.7. Experimental Methods for Studying the Gas Flow Pattern

**5.7.1. Measurement of the Local Gas Velocity in a Furnace.** The pitot tube or any other method of measuring gas velocity at a point, discussed in Section 5.4.4, can be used for measuring the distribution of velocity across a stream. In studying turbulent flow this type of measurement gives the mean velocity; if the measurement is made in a region of "eddy" flow, the readings fluctuate considerably in time and this provides a simple way of discovering the presence of such regions. The use of pitot tubes for the construction of a complete picture of the flow pattern is, however, laborious and hence the method does not lend itself to the rapid verification of the effect of changing the system.

Since mixing and combustion in diffusion flames are so closely connected (see 3.3.3) the measurement of gas composition at points across a furnace in which separately introduced gas and air streams are burning can be used as a simple measure of mixing. The use of the water-cooled multiple gas-sampling probe described in Section 3.3.4 is therefore one of the best experimental ways of investigating the relative mixing produced by different burners or port designs.

**5.7.2. Methods of Making Flow Visible.** In experiments on cold systems such as aircraft in wind tunnels very valuable results can be attained by using a transparent wall and arranging smoke in the gas\*. In water models suspended particles of aluminium or small air bubbles when suitably illuminated can be used for studying details of flow.<sup>5.34, 5.35</sup> These methods are, unfortunately, not possible for hot furnaces, where it is very difficult not only to give the gases different relative visibility but also to obtain sufficient sight-holes. In certain special cases, however, the flow can be made visible. Thus to study the flow in an air regenerator, a mixture of aluminium and soot has been found useful. The mixture is inserted by means of an iron spoon through a hole made by removing a brick at the base of the regenerator and its subsequent track in the regenerator can, with care, be traced by the

\* An excellent smoke for cold flow experiments is provided by permitting air saturated with hydrochloric acid to come into contact with fumes of ammonia.

<sup>5.34</sup> CHESTERS, J. H., HOWES, R. S., HALLIDAY, I. M. D., and PHILIP, A. R.: "Open-hearth Furnace Models," *J.I.S.I.*, **162**, Pt. 4, p. 385 (Aug. 1949).

<sup>5.35</sup> NEWBY, M. P.: "Experiments on the Gas and Fluid Flow in a Side-blown Converter Model," *J.I.S.I.*, **162**, p. 452 (Aug. 1949).



appearance of sparkles in the air when the regenerator is viewed from similar holes higher up. Wooden blocks supported by iron rods have also been used<sup>5.36</sup> to render the flow in furnace systems visible by the streaks of luminous flames given off by the burning wood. Cooper and Beral<sup>5.37</sup> have described a water-cooled periscope which can be used to take wide-angle photographs of conditions inside a furnace through a small hole. The use of this periscope in conjunction with various methods of making flow visible will clearly be of great value in further furnace studies.

**5.7.3. The Radon Technique.**<sup>5.16</sup> Radon gas can be used for measuring the type of flow in a furnace and the mean mixing length, its function being analogous to that of the dye used in water tests. The radon is inserted into the flow at a certain point by means of a water-cooled tube and the distribution of concentration is measured across the stream at subsequent points in the flow. This is done by withdrawing samples of the gas from different points and measuring their radioactivity in an electroscope; the concentration is then plotted as a function of the distance perpendicular to the direction of flow at various distances downstream. This gives a picture of the rapidity of mixing of the gas at the point of entry of the radon with its surroundings and also allows the mean streamline through this point to be determined, since the maximum concentration of radon will lie on this line. This method has the advantage over the methods discussed in Section 5.7.2 that a single hole suffices for the insertion of a probe to all the points across the cross-section and thus gives a complete flow picture in this section. It is also possible to derive the mean mixing length from the width of the spread curve at various points downstream.

**5.7.4. Models.** Both cold air and water have profitably been used in small-scale models for studying the subdivision of gas flow in furnace systems, and this type of work will become of greater and greater value as designers begin to use it to test each new plan. The use of air is in general not so valuable for visual observation of the flow pattern because the change in viscosity from the original is not great enough to slow the flow down sufficiently for easy visibility. Thus in the case discussed in Section 5.3.3 of air models for resistance measurements, the air takes only as long to traverse the

<sup>5.36</sup> CHESTERS, J. H.: Private communication.

<sup>5.37</sup> BERAL, L., and COOPER, W. T.: "Photographic Techniques in Combustion Research," *Fuel*, 27, No. 1, p. 10.



length of the furnace model as it does to traverse the corresponding length which is 12 times greater in the original. Water at 15° C. has a kinematic viscosity about  $\frac{1}{13}$  of that of air at this temperature, and hence for a model of the same size at the same Reynolds number the velocities are  $\frac{1}{13}$  as great as with air at 15° C. This in general brings the time scale to a value suitable for studying with the eye, and hence greatly increases the usefulness of the model for assisting designers in visualising what is happening, which is one of the most useful aspects of model work. Air models are, however, quicker to make, set up and alter than water models and it is likely that they will continue to find a use in subdivision problems when coupled with techniques such as flash photography, high speed cinematography and stroboscopy. A method of using air models for the evaluation of mixing rates similar to the radon method for furnaces has recently been developed<sup>5.28</sup> based on the infra-red gas analyser. In this method cold air is used to represent the flow of preheated air, producer gas and combustion gases, and a concentration of about 0.03% CO<sub>2</sub> (additional to that already present in both streams) is introduced into the air stream which simulates the producer gas at a point prior to its entry into the furnace so that it is fully mixed with this stream when it enters. Then the ratio of the concentration  $C_1$  of CO<sub>2</sub> (above normal surrounding value) in the mixture of the two streams which occurs at any point in the furnace to the concentration  $C_0$  in a sample of the entering "gas" stream, is equal to the ratio of the "gas" to gas and air in the mixture. The samples were taken out with a fine tube connected via the infra-red gas analyser to a suction fan and a reading could be obtained of the CO<sub>2</sub> concentration in 15 seconds. Since mixing is so important in diffusion combustion the use of this method to evaluate mixing in models is clearly of great value. Moreover, air models are much more readily adapted to small changes of design than are water models, so that this method has much to commend it for use by designers. The air model can even be used with reasonable approximation to study the mixing of air with a steam or air-borne jet of oil droplets. A typical model of an open-hearth steelmaking furnace is shown in Fig. (5)19. This model,<sup>5.35</sup> which is about 3 ft. long and is constructed to  $\frac{1}{24}$  scale, was used for studying the pressure losses in all the various parts of the furnace system and for obtaining the mean velocity profiles of the flowing gases across a number of different sections of the working chamber.

Models based on the principle of similarity using water flow, such



as those used by Groume-Grjmailo and, more recently, by Rosin,<sup>5.38</sup> Chesters<sup>5.34</sup> and Newby,<sup>5.35</sup> have likewise a considerable field of application for studying the subdivision of flow. Fig. (5)20 shows a photograph of the flow which occurs in a model of a side-blown converter used for steelmaking. Water is used to simulate the air blown through the tuyeres on the full-scale converter and mercury in the bottom of the chamber simulates the molten metal. The water is entering on the bottom left of the model and the

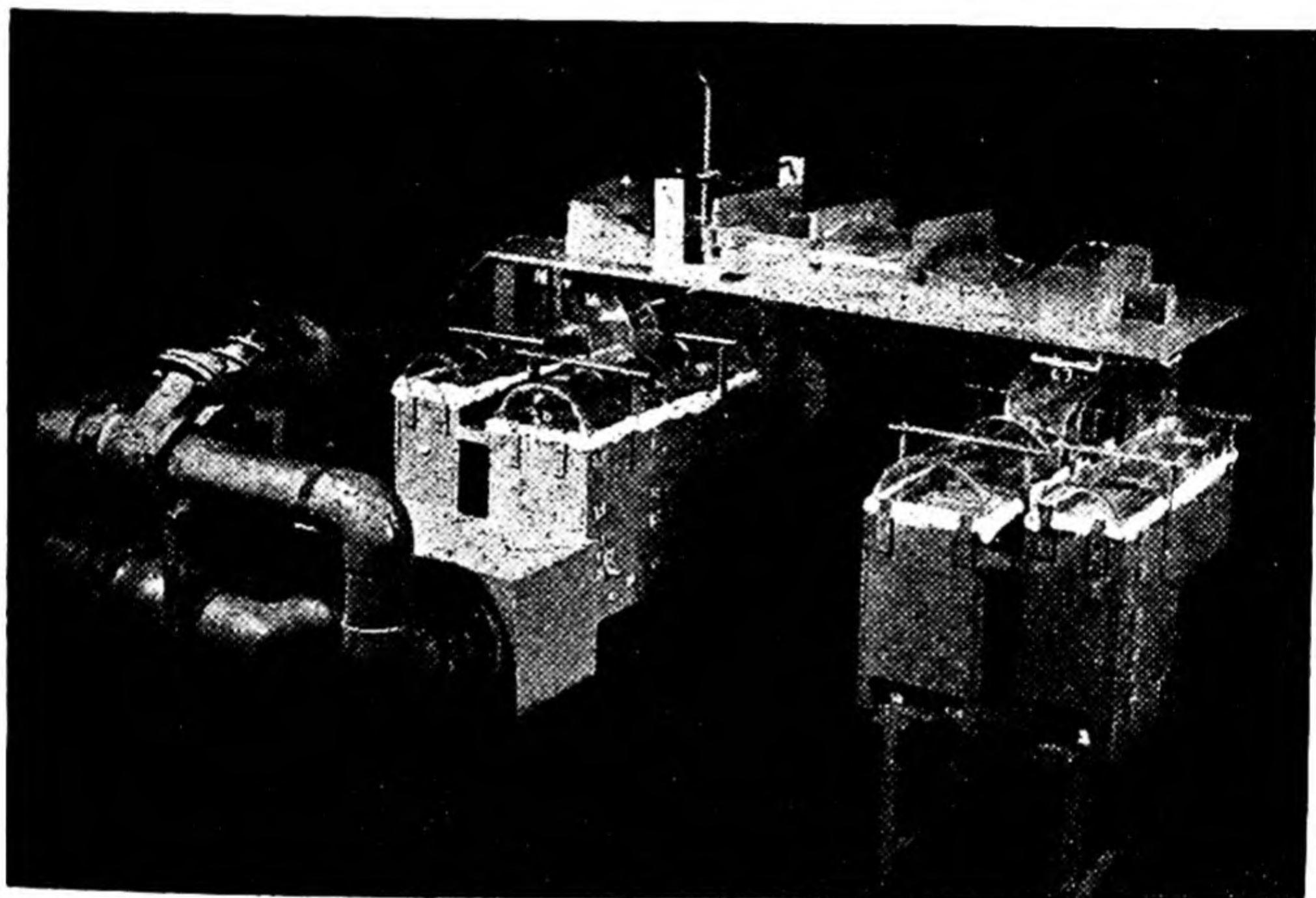


Fig. (5)19. Aerodynamic model of Open-Hearth Furnace for studying the influence of design upon flow conditions.

greater part of the chamber is seen to be filled with a recirculating flow.

Visualisation of the flow is given by small air bubbles which are illuminated by a narrow sheet of light.

Models are, however, limited by (1) the fact that it is only possible to obtain a Reynolds number identical with that in the furnace at one chosen point, since the viscosity of the furnace gas changes along the system, whereas that of the water or air used in the model does not. (2) No exact analogy with the effects of buoyancy due to the variation of density within the flowing system has been

<sup>5.38</sup> ROSIN, P. O. : *J. Inst. Fuel* (1938).



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attained. Two attempts have been made to introduce a buoyancy effect. Groume-Grjmailo used two fluids of different density, but failed to take account of the mixing. Rosin's method involves the solution of salt briquettes by the water, the model being turned upside down so that the increase of density due to the saturation of

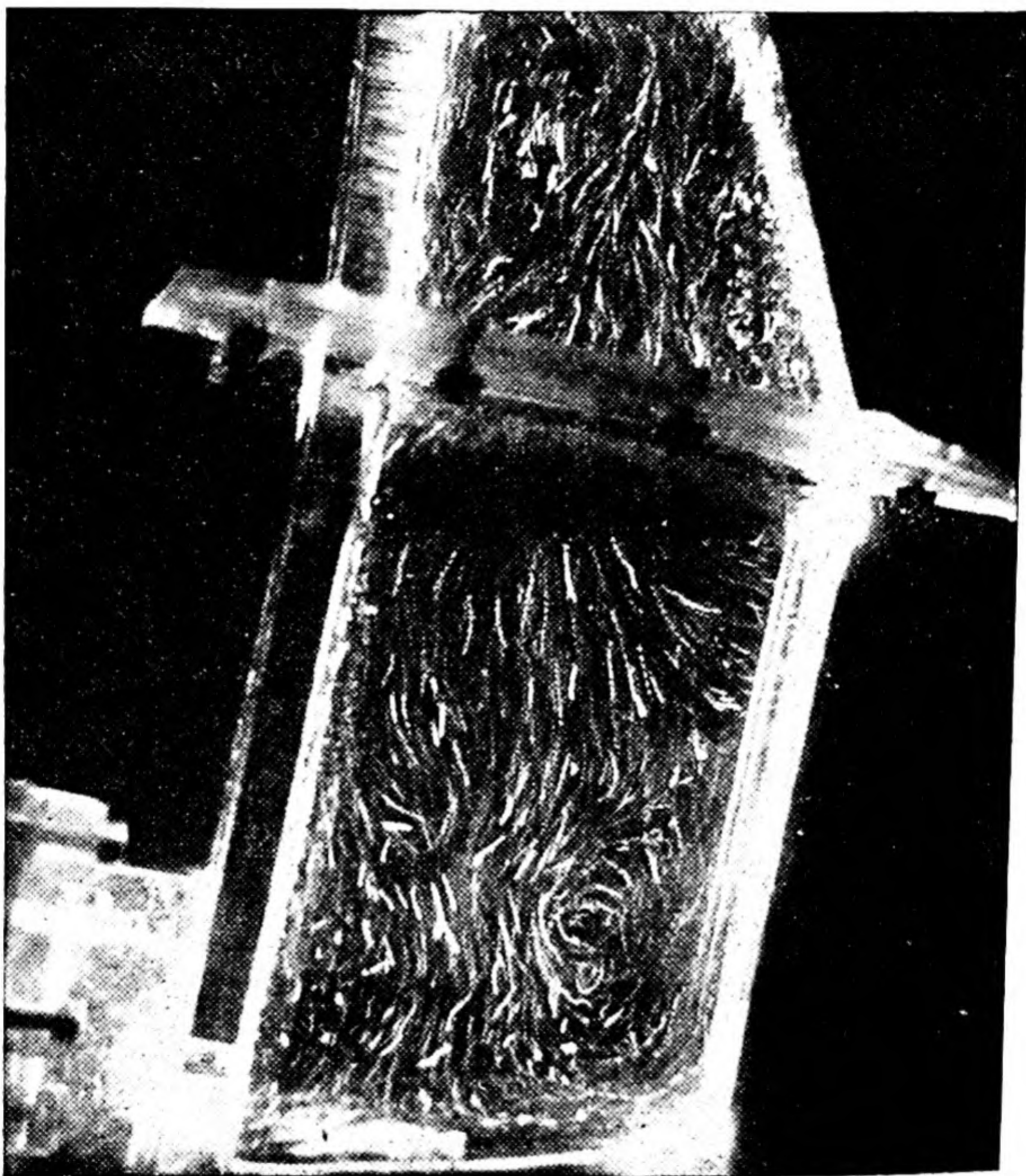


Fig. (5)20. Flow in a model Side-Blown Converter operated with water.

the water by salt is partially analogous to the rise of temperature when the gas is burned with solid fuel. This is open to the criticism that the change in density is only a rise of about 40%, whereas in combustion the density of the burning gases may fall to  $\frac{1}{3}$  or  $\frac{1}{4}$  of the cold value, but it does give a good analogy to buoyancy. (3) It has so far not been found possible to incorporate the effects, of air inleakage described in Section 5.4.6 in a model.



For these reasons the similarity between models and actual furnaces is never perfect. On the other hand, they do give a general picture of the flow conditions, both as regards subdivision of flow across the stream and as regards the turbulence, breaking-away of flow and mixing length, which is of great use to designers. Hence the construction of models is, in many cases, amply repaid by the results obtained. It is recommended that in all cases where a large number of similar furnaces have to be built, a model should be constructed and the furnace designers should spend some hours watching it in order to train the imagination in the flow processes. Examples of cases where models have been of great use are the regenerator or a glass tank, the gas-producer and the heating flues of coke ovens. A further advantage of the use of models is that it becomes a matter of minutes to alter the shape of the system and see the effect of this alteration on the flow pattern. This means that suggestions can be tried which would be too numerous or too drastic to be attempted directly in actual furnaces. Those which the model shows to be promising can then be built.

Water furnace models may be constructed of a suitable transparent plastic about  $\frac{3}{16}$  inch thick. They are connected by flanges to a water flow system, either with recirculation or with discharge to waste. The water quantity is regulated and measured by means of an orifice to obtain the desired Reynolds number. The flow is made visible by introducing dye through a tapered glass tube inserted through a rubber stopper in the wall of the model and fed from a bottle at a suitable height. When the flow velocities are low the dye should be adjusted by admixture with alcohol to give it exactly the density of water. In strongly turbulent flow it is usually better to introduce the dye in short pulses, as the mixing is so great that the whole system becomes darkened if it is introduced continuously.

Some experiments have also been made on the flow of cold air in full-scale furnaces (glass-tank<sup>5.1</sup> and open-hearth furnace<sup>5.39</sup>) before the latter have been heated up or after they have been cooled. It is then possible for observers to enter the furnace and observe the details of the flow pattern provided Reynolds number similarity compared to the hot furnace can be obtained by drawing enough cold air through with a blower fan or suction fan. This has the advantage over small-scale models that perfect geometrical

<sup>5.39</sup> NEWBY, M. P., British Iron and Steel Research Association Report SM/A/134/50.



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similarity is automatically obtained, all roughness of surfaces, etc., being taken into account. Also it is unnecessary to build a special model, so that results can be quickly obtained when a furnace is built. On the other hand, all the other failures of similarity which occur in small models also apply to cold tests on full-scale ones, and it is not so easy to alter the system on the full scale. For these reasons the two methods each have a part to play in improving the flow patterns in furnaces.

## SUMMARY OF CHAPTER 5

### THE AERODYNAMICS OF HOT SYSTEMS

#### 5.1. The Factors governing the Total Quantity of Gas flowing in a Circuit

**5.1.1. The Significance of High-temperature Aerodynamics.** Combustion and heat transfer are strongly affected by gas flow. Gas flow in hot systems is the resultant of viscosity (fluid friction) inertia and aeromotive forces (buoyancy and fans).

**5.1.2. The Two Basic Problems of High-temperature Aerodynamics.** To design furnaces it is necessary to know (1) how to calculate the total quantity of gas that will flow through the system, and (2) how to predict the pattern of the gas flow inside specific parts of the system (e.g. velocity distribution).

**5.1.3. The Laws Governing the Total Quantity Flowing in a Channel.** The general law governing the quantity of gas flowing in a system of channels is

$$\Delta p = \Delta p_B + \Delta p_K + \Delta p_R \quad . \quad . \quad . \quad (5/3)$$

$\Delta p_K$  is pressure change due to change of dynamic head and is given by the formula :

$$\Delta p_K = \frac{\rho_1 \rho_2 (v_1^2 - v_2^2)}{\rho_1 + \rho_2} \text{ approximately}$$

if  $\rho_1 = \rho_2 = \rho$

$$\Delta p_K = \frac{\rho (v_1^2 - v_2^2)}{2}$$

where  $\Delta p$  is the overall change in total pressure referred to atmospheric at the same height at each end ;

$\Delta p_B$  is the buoyancy pressure (or other aeromotive force) =  $(\rho_o - \rho)gh$  ; and

$\Delta p_R$  is the resistance to flow which is a function of the quantity of gas flowing (see 5.3).



**5.1.4. The Electrical Analogy.** It is useful in describing gas flow to use an analogy with electric currents ; fans and buoyancy are sources of aeromotive force similar to the battery, and the gas flow resistance is equivalent to electric resistance.

**5.1.5. Turbulent and Streamline Flow.** The flow of gases in furnaces is nearly always turbulent, the state of flow being characterised by the Reynolds number  $R_e = vD/\nu$ , which always exceeds the critical value. In some parts of the system the turbulence is steady, while behind obstructions it breaks away from the surface and is "eddying."

## 5.2. Total Flow in a Circuit. Sources of Aeromotive Force

### 5.2.1. Buoyancy

$$\Delta p_B = 0.0147h \left( \frac{273}{273 + T_a} - \frac{273}{273 + T_s} \right) \frac{\rho_s}{\rho_a} \quad . \quad . \quad (5/14)$$

where  $h$  = vertical height of first point above second ;

$T_a$  = temperature of atmosphere ;

$T_s$  = mean temperature of furnace gases in column ; and

$\frac{\rho_s}{\rho_a}$  = ratio of density of furnace gases at N.T.P. to density of air at N.T.P.

**5.2.2. Fans.** Where fans are used to provide aeromotive force, the designer can specify the value of the aeromotive force required and obtain a fan to give this value.

**5.2.3. Injectors.** A method of calculating injectors is given based on the law of conservation of momentum.

## 5.3. Total Flow in a Circuit. Resistances

### 5.3.1. Friction Loss in Long Channels

$$\frac{\Delta p_R}{\rho v^2 / 2g} = 0.187 \times F \times \frac{L}{D} \quad . \quad . \quad . \quad (5/18)$$

where  $F$  is dimensionless friction factor which is independent of the Reynolds number  $R_e$  for high values of  $R_e$ , but depends very much on the pipe roughness.

### 5.3.2. Pressure Drop at Bends and Changes of Section

$$\frac{\Delta p_R}{\rho v^2 / 2g} = 0.0156 \times K \quad . \quad . \quad . \quad (5/19)$$



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**5.3.3. Resistance of Beds of Solid Particles.** No exact formula can be given, since the resistance depends in an irregular way on packing.  $\Delta p_R$  is given between limits of 4 : 1 and  $\frac{1}{4}$  : 1 by

$$\frac{\Delta p_R}{\rho v^2} \cdot \frac{\epsilon^3}{LS_1} = 0.15 \left( \frac{vS}{v} \right) + 0.012 \left( \frac{vS}{v} \right)^{0.1} \quad . \quad . \quad (5/20)$$

### 5.3.4. Resistance of Regenerators

$$\frac{\Delta p_R}{\rho_O v_O^2} = \frac{C.T.H.}{d^{0.25}} \quad . \quad . \quad . \quad . \quad . \quad (5/21)$$

where  $C = 5.1 \times 10^{-6}$  for straight-through checkers and  $7.1 \times 10^{-6}$  for double-staggered checkers.

**5.3.5. Resistance of Furnace Walls to Leakage.** Leakage  $Q$  in ft.<sup>3</sup>/sec. given by

$$Q = \frac{\alpha \cdot A \cdot \Delta p}{d} \quad . \quad . \quad . \quad . \quad . \quad (5/22)$$

where  $\alpha = 0.16$  for very good mortar joints, 5.6 for dry joints ;  
 $A$  = area of wall in ft.<sup>2</sup> ; and  
 $d$  = thickness of wall in inches.

## 5.4. Measurement of the Total Quantity of Gas Flowing in a Circuit

**5.4.1. Measurement by the Pressure Differential across a Restriction. Theoretical Basis.** From Bernoulli's theorem the static pressure  $p_1$  in a restriction should be less than that of  $p_2$  in the main tube by

$$p_2 - p_1 = \frac{\rho Q^2}{2} \cdot \frac{1 - m^2}{m^2 A_2^2} \quad . \quad . \quad . \quad . \quad (5/23)$$

if  $m$  = area ratio of the restriction ;

$Q$  = quantity of gas flowing,  $\rho$  is its density ; and

$A_2$  = area of main tube.

For practical systems the measured pressure difference is equal to this value multiplied by a factor  $CZ$ .

**5.4.2. Calculation of Orifices and Venturis.** Details are given of the method of calculating the quantity of gas flowing when the pressure difference and the orifice diameter are known, and of the method of calculating the required orifice diameter when the flow quantity and pressure difference are known.

**5.4.3. Special Uses of Orifices and Restrictions.** The restriction method of measuring flow quantities can be used even where the proper dimensions cannot be adhered to, provided the system is calibrated. Provided the velocity in the orifice is high, orifices



with open chambers on one or both sides can be used without calibration.

**5.4.4. Measurement of Quantity by Measurement of Velocity at Points on a Cross-section.** The quantity of gas flowing can be measured by means of velocity measurements if the cross-sectional area of the channel is subdivided into a large number of equal areas and the velocity in the centre of each of these is measured. For measurement with the pitot tube :

$$v = \frac{18.29}{\sqrt{\rho}} \cdot \sqrt{h} \text{ ft./sec.} \quad . \quad . \quad . \quad (5/28)$$

**5.4.5. The Dilution Method.** Gas quantities can also be measured by measuring the concentration obtained when a detectable impurity is mixed in at a known rate. One way is by the use of radon gas.

**5.4.6. Measurement of Leakage.** The quantity of air leaking into a furnace system can in some cases be deduced from gas analyses at different points. In some other cases it can be measured by a pitot tube.

## 5.5. Total Flow in a Circuit. Representation and Summary of Results

**5.5.1 Pressure Balances for Proposed Furnaces.** Using the results of 5.2, 5.3 and 5.4, the complete flow calculations of a furnace can be constructed and hence the proposed dimensions checked.

**5.5.2. Pressure Balances for Furnaces which are already Built ; Comparison of Calculated and Measured Pressures.** Measurement of the pressures on a furnace enables the cause of deviations from expected aerodynamic performances to be traced.

**5.5.3. Graphical Pressure Balances.** A method of representing the results of pressure measurements of a furnace system which brings out clearly the changes along the gas path is given in Fig. (5)14.

## 5.6. The Pattern of Gas Flow

**5.6.1. Stratification of Flow in Horizontal Streams.** The fact that the cool charge is on the floor tends to create a stagnant pocket in contact with the charge. The use of high gas velocities reduces this effect.

**5.6.2. The Path of a Jet.** A jet issuing from a nozzle (of diameter  $d$ ) into a furnace chamber loses velocity by entraining the surrounding gases. If its temperature differs from that of the



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surrounding gases it is also deflected to the action of buoyancy, while any superimposed velocity of the surrounding gases will likewise cause a deflection of the axis of the jet.

**5.6.3. The Effect of Buoyancy on the Distribution of Gas Flow when the Flow Direction has a Vertical Component.** If gases flow downwards when they are hotter than their surroundings, buoyancy will tend to make them uniformly distributed across a horizontal section, and conversely.

**5.6.4. The Speed of Mixing of the Two Streams.** In turbulent flow, mixing takes place by the movement of eddies across the stream, and the turbulence may be characterised by a mean movement of such eddies.

**5.6.5. The Speed of Entrainment of the Surrounding Fluid by a Jet.** The velocity at any point in a free jet distant  $x$  along the axis and  $y$  from the axis is given by

$$\frac{u}{u_0} = \frac{1}{(1 + 64y^2/x^2)^2}$$

where  $u_0$  is the velocity on the axis for the same  $x$ ;  $u_0$  is related to the jet momentum  $G$  by

$$G = 0.0164 \rho u_0^2 x^2 \text{ ft.-lb./sec.}^2$$

and the mass of jet plus entrained air at distance  $x$  is given by

$$M = 0.38x \cdot \sqrt{G \cdot \rho}$$

In these formulæ  $x$  is measured from a distance  $0.6d$  back from the jet pipe mouth.

Self-aggravating irregularities may appear, however, due to combustion or to high rates of flow which disturb the particles composing the bed.

**5.6.6. Flow Distribution across Beds of Broken Solids.** Where the packing is perfectly regular the bed acts as a very good flow distributor, completely eliminating irregularities.

## 5.7. Experimental Methods for Studying the Gas Flow Pattern

**5.7.1. Measurement of Local Gas Velocity.** The simplest way of building up a flow picture is by the use of pitot tubes to measure the velocity distribution.

**5.7.2. Methods of Making Flow Visible.** In some cases the flow in furnaces can be made visible by the use of indicators such as aluminium dust.



**5.7.3. The Radon Technique.** Radon can be used as an indicator of the flow pattern by introducing it at one point and measuring the distribution across the flow downstream.

**5.7.4. Models.** It is difficult to obtain complete similarity with water models, but nevertheless they are frequently of great value in furnace design.



## 6. THE SCIENCE OF FURNACE CONSTRUCTION

### 6.1. The Furnace Skin

The previous chapters have been concerned with the scientific principles which fix the design of the furnace. On the other hand this chapter deals with (a) the extent to which that design is permitted by the materials available without collapse, fusion or other types of failure under working conditions ; and (b) the principles of construction which lead most nearly to the achievement of the desired performance with the materials available.

We may begin with the consideration of what would be the properties of an ideal furnace structure. Clearly such a structure would have to *retain its shape* during operation; hence the refractory must be volume stable. This ideally involves (i) good spalling resistance\* if the temperatures fluctuate in operation, and (ii) at maximum operating temperature, zero shrinkage or expansion even under stress. In addition *chemical inertness* to gaseous, liquid and solid environment is essential. Finally the structure must *retain all the heat and gases* so that minimum thermal conductivity, specific heat and gas permeability are required. It should in fact completely isolate the high-temperature region inside the furnace from the cold surroundings; the unavoidable exceptions being apertures through which the raw materials enter and the products and waste materials leave the furnace. The only other communications, between the hot furnace and the cold surroundings are those by which the process is controlled.

While the ideal refractory is always required to be volume stable and chemically inert, there are some parts of furnaces in which the other properties should be different from those outlined above. In muffles or recuperators high conductivity is required. In regenerators, maximum specific heat and thermal conductivity are desirable. An unusual furnace described in a recent paper<sup>6.1</sup> requires maximum permeability in certain refractories because the waste gases are taken off through the walls to which the gases give up their heat.

\* This may be roughly defined as the tendency for the bricks to crack through when the heat flow to the hot face is suddenly changed.

<sup>6.1</sup> ANDERSON, R. H., GUNN, D. C., and ROBERTS, A. L. : "Uses of Permeable Refractories for Furnace Construction," *J. Inst. Fuel*, 17, p. 169 (Aug. 1944).



Returning to the requirements for the "skin," all actual refractory materials have in fact a finite thermal conductivity,<sup>6.2</sup> so that some heat always escapes through the walls.\* As will be shown in this chapter, it also works out that those refractories which are suitable for use at the highest temperatures have a relatively high thermal conductivity. Consequently in many cases either no low-thermal-conductivity brick is used at all, as in most open-hearth furnaces, or else the furnace skin is made not by a single layer of brick, but by a series of layers<sup>6.3</sup> each one thick enough to bring the interface temperature at its outside down to that which can be withstood by its successor (the calculation of such an arrangement has been discussed in Section 4.4.1) and the inner layers usually have to be considerably thicker than the outer ones in order to satisfy this condition, owing to their higher thermal conductivity. In practice the inner layers are often made even thicker than corresponds to this condition in order to ensure mechanical stability and allow for wearing away. Also, in some types of industrial furnace it is the custom to operate the furnace at a temperature equal to that which can just be withstood by the innermost refractory so that it is necessary to have a fairly large heat flow through the wall in order to make the inner refractory surface cooler than the furnace,<sup>†</sup> and, still more important, ensure that there is a large temperature gradient through it so that any fluxing of the inner surface does not penetrate to an important depth. For example, it is for this reason that, in this country, the silica crowns of open-hearth steel-making furnaces are very rarely insulated (although this insulation has on occasion been successfully carried out<sup>6.4</sup>). The insulation caused by accumulated dust on electric arc furnace roofs has been found to diminish the roof

\* The method of calculating this heat loss through the walls has been described in Section 4.4, while in 7.4.3 a method of evaluating the effect of this heat loss upon the overall heat usage by determining the variation of efficiency with output is given.

† This effect has been shown to be very small in *I.S.I. Special Report* No. 37, p. 175, where it is demonstrated that even for an uninsulated 9-inch silica brick roof the net heat conducted through the roof is only about 1% of the gross radiation falling on the inside surface.

<sup>6.2</sup> NORTON, F. H.: "The Thermal Conductivity of some Refractories," *J. Amer. Cer. Soc.*, **10**, p. 30 (1927).

<sup>6.3</sup> (a) "The Insulation of Furnaces," Ministry of Fuel and Power, Fuel Efficiency Bulletin No. 17 (July 1943).

(b) ETHERINGTON, R.: *Modern Furnace Technology* (Griffith, London, 1938).

<sup>6.4</sup> HARVEY, F. A.: *J. Amer. Ceram. Soc.*, **18**, p. 86 (Mar. 1935).



## 6.1 THE SCIENCE OF FLAMES AND FURNACES

life, <sup>6.5</sup>, <sup>6.6</sup> but there the conditions are even more arduous than in an open-hearth furnace.

As regards the tendency to spalling, this is reduced in practical furnace design by building the walls of pre-fired bricks which are small compared with the size of the wall, so that expansion can take place along the joints; or, alternatively, of unfired material rammed *in situ* to form a monolithic structure, *i.e.* which has a less uniformly rigid structure. There is, however, a greater tendency for monolithic linings to crack by contracting at high temperature. In spite of this they are very successful in many furnace bottoms and as walls in small furnace units.<sup>6.7</sup> Furnace walls made with bricks are, as has been shown in Section 5.3.6, always liable to have gas leakage. In the case of the external skin this can be overcome in most parts of the structure by the application of external metal casing and of doors sloping so that they fall tightly against their frames, although such methods are not used to anything like the extent that the importance of air infiltration would justify. As far as the inner partitions are concerned (see below, Section 6.4) it is extremely difficult to obtain satisfactory gas-tightness at high temperatures.

## 6.2. Tests for Refractories

**6.2.1. The Purpose of Refractory Tests.** In Section 6.1 it has been shown what would be the properties of an ideal refractory for furnace construction and that actual refractories always deviate more or less from this ideal. It is consequently necessary to be able to measure the properties of each refractory by small-scale laboratory tests. Such tests are, however, necessarily of limited value, since they can never take account of all the conditions of actual practice.\* Nevertheless, a number of tests have been very extensively and carefully developed which do give guidance on the suitability of a refractory for a particular application.<sup>6.8</sup>, <sup>6.9</sup> In the next three sub-sections therefore such tests will be classified accord-

\* The same objection applies to *all* laboratory tests intended to throw light on furnace operation.

<sup>6.5</sup> PASCHKIS, V.: *Industrial Electric Furnaces and Appliances*.

<sup>6.6</sup> PASCHKIS, V.: "Arc Furnace Output," *Iron and Steel*, **120**, (1947).

<sup>6.7</sup> First and Second reports on the Development of Monolithic Dolomite Linings: *I.S.I. Special Report*, No. 33 and 35.

<sup>6.8</sup> See especially J. H. Chester's *Steelplant Refractories* (United Steel Companies).

<sup>6.9a</sup> Institution of Gas Engineers Standard Specification for Refractory Materials, Publication No. 301 (1946).

<sup>6.9b</sup> Tentative Standard Methods of Testing Refractory Materials, *B.R.R.A. Special Publication* No. 8 (May 1946).



ing to the three requirements of an ideal refractory, namely, (1) dimensional stability under working conditions ; (2) appropriate thermal conductivity ; and (3) resistance to chemical attack.

**6.2.2. Dimensional Stability Under Working Conditions.** (a) *Melting-point.* The most obvious requirement under this head is that the refractory should not melt at the desired temperature. One way of assessing this property is the cone test,<sup>6.10</sup> in which a cone 1½ inches high and with a ¾-inch base is cut from a sample of brick and is surrounded by standard smaller cones of graded composition. The group is heated in a furnace at fixed rate of heating (10° C. per minute) and the heating is discontinued when the tip of the sample is bent over level with the base. The temperature at which this takes place is estimated both by noting which of the standard cones has just bent over to touch the base and also in many cases by an optical pyrometer. The first of these two temperatures is known as the pyrometric cone equivalent (P.C.E.). This test should be carried out in air, as variations in the oxygen partial pressure can produce entirely different minerals and consequently melting temperatures in some refractories.

(b) *Refractoriness under load.* In actual practice, refractories must be capable of withstanding not only their own weight but additional compression, due for example to the weight of the bricks above them or the thrust of a crown (see Section 6.4 below for the evaluation of this thrust). Neither the pyrometric cone equivalent nor the final melting-point give the temperature at which a refractory will collapse under load. This is because some refractories are complex and melt over a range of temperature, while others are pure compounds with a sharp melting-point; and because such factors as type of crystal structure and the viscosity and amount of the liquids formed also affect the behaviour under load. In some cases also, impurities may be concentrated near the surface of the crystals. These surfaces may therefore soften and act as a lubricant at a temperature far below the melting-point of the major constituent.

As a consequence of these differences in behaviour, it is necessary to develop a test for bricks which will apply stresses corresponding more closely to those the bricks will encounter at high temperatures in actual furnace practice. For this purpose various refractoriness-under-load tests have been developed. In all cases the specimen,

<sup>6.10</sup> *A.S.T.M. Manual of Refractory Materials*, 1946, c. 24-46. FAIRCHILD and PETERS : *J. Amer. Ceram. Soc.*, 9, p. 701 (1926). See also <sup>6.8</sup>, <sup>6.9</sup>.



## 6.2.2 THE SCIENCE OF FLAMES AND FURNACES

which in British methods is  $3\frac{1}{2} \times 2 \times 2$  inches<sup>6.8, 6.9, 6.11</sup> and in an American method a full-size brick  $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$  inches, is heated evenly on all sides, and loaded in the end-on position. The loads used are 25, 28<sup>6.12</sup> or 50 lb./in.<sup>2</sup>, and the change in length is observed.

There are two types of test, the rising-temperature test<sup>6.8</sup> (Fig. 6(1)) used in Britain in which the specimen is heated at 10° C. per minute, which underrates the effect of time, and secondly the maintained-temperature test.<sup>6.9</sup> A variation of the latter is J. H. Partridge's creep test.<sup>6.13</sup> To give a single criterion of performance

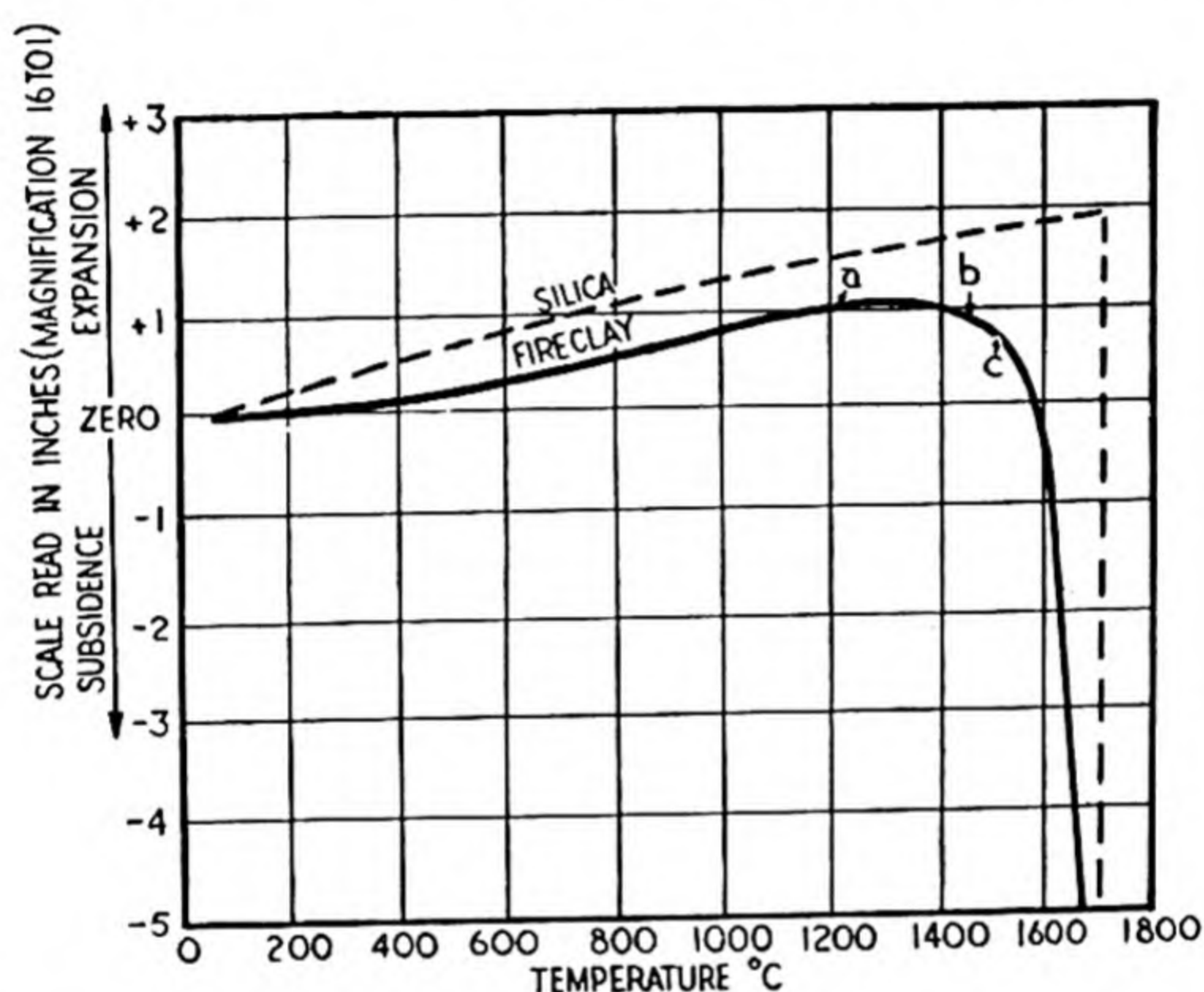


Fig. (6)1. Typical rising temperature test curve for refractoriness under load.

instead of the complete underload curve the results may be expressed in various ways. In the rising-temperature test the point of failure is taken as the temperature at which 5%, or in some tests 10%, subsidence has taken place. Results of a maintained-temperature test are expressed as the amount of subsidence occurring in a stated time at a given temperature.

The fact that in practice bricks are rarely subjected to anything like these loads when heated right through results in tests having a fairly large margin of safety when applied to walls or arches which are cold on the outside, even though they may be loaded to a total

6.11 MELLOR and MOORE : *Trans. Cer. Soc.*, **15**, p. 117 (1916).

6.12 Deutsche Industrie Normen.

6.13 PARTRIDGE, J. H. : "The Testing of Glass House Pots," Pt. IV, *J. Soc. Glass Tech.*, p. 190 (1931) ; also NORTON, F. H. : *J. Amer. Ceram. Soc.*, **22**, 16, p. 334 (1939).



loading of about 30 lb./in.<sup>2</sup> in extreme cases. Refractories are rarely subjected to tensional and torsional applied stresses, but tests have been developed for determination of the behaviour under both these types of stress.

(c) *Permanent contraction or expansion.* Some refractories are used at temperatures which are high enough to produce in them a permanent expansion or shrinkage. This behaviour is measured as for refractoriness under load, on test-pieces  $3 \times 2 \times 2$  inches or full bricks  $9 \times 4\frac{1}{2} \times 2\frac{1}{2}$  inches ; these are heated for a standard time at a given temperature, depending on the refractory, and re-measured.<sup>6.14</sup> The percentage expansion or contraction is derived from these measurements.

(d) *Spalling, cracking and expansion.* The reversible thermal expansion of refractories is measured<sup>6.2</sup> by a straightforward measurement of the change in length in an electric furnace. In order to allow for the change in length of the connecting pieces which pass from the high-temperature region to the measuring apparatus, it is usual to have a standard test-piece of known thermal expansion alongside, with identical connecting pieces, for the purpose of calibration.<sup>6.15</sup>

Thermal expansion alone, however, is not a sufficient test for the behaviour of refractories on rapid heating and cooling, since the thermal conductivity and strength also come in. The tendency of refractories to crack and portions to fall off when they are heated and cooled rapidly, is known as spalling, and two of the various direct tests for the relative spalling properties of different refractories are as follows.

Chesters<sup>6.16</sup> describes a test on a single sample  $3 \times 2 \times 2$  inches. This is heated at 900° C. in a furnace of sufficient heat capacity not to be appreciably cooled by the sample. After 10 minutes it is removed and placed on a brick floor, where it is allowed to cool for 10 minutes. This heating and cooling cycle is repeated until the brick is in such a condition that on cooling it can be readily

<sup>6.14</sup> *A.S.T.M. Manual of Refractory Materials*, c. 113–36. VICKERS, A. E. J., and SUGDEN, J. A. : *J. Soc. Col. Tech.*, **17**, p. 320 (1933).

<sup>6.15</sup> (a) HODSMAN, H. J., and COBB, J. W. : *J. Soc. Glass Tech.*, **3**, p. 201 (1919).

(b) MERRITT, G. E. : *U.S. Bureau Stds. Sci. Paper*, **19**, p. 357 (1924).

(c) MELLOR and MOORE : *Trans. Brit. Cer. Soc.*, **15**, p. 117 (1916).

(d) HYSLOP, J. F., PROCTOR, R. F., and BRIGGS, H. C. : *J. Soc. Glass Tech.*, **12**, p. 190 (1928).

(e) PARTRIDGE, J. H. : "The Testing of Glass House Pots," Pt. IV, *J. Soc. Glass Tech.*, p. 190 (1931).

<sup>6.16</sup> CHESTERS, J. H. : *Iron Age*, pp. 47–51 (13 Feb. 1941).



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pulled apart in the hands. Loss of corners or small cracks are recorded, but actual failure is not regarded as having taken place until the test-piece can be pulled into two approximately equal pieces. A brick which withstands more than thirty cycles is regarded as excellent from this point of view.

It is certain, however, that a test on a single brick cannot even approximately reproduce the conditions of spalling in actual furnaces. For this reason panel tests have been developed where a number of bricks are built into a panel which is heated, like the wall of a furnace, on one face only and then cooled a number of times, the resulting damage being noted. An example of such a test is given by Hyslop and Briggs.<sup>6.17</sup> The American (A.S.T.M.) spall test<sup>6.18</sup> is of this type and is generally accepted. Spalling resistance and resistance to chemical attack are to some extent mutually exclusive, as the former requires, as a rule, a high porosity and the latter a low one: usually 18–27% porosity is the optimum range for general-purpose bricks. Exceptions are refractories made of materials with a very high conductivity, such as fused alumina. In this case it is possible to combine low porosity with good spalling resistance.

**6.2.3. Thermal Conductivity.** In the general method of thermal conductivity measurements, a flat panel consisting of the test-piece surrounded by a fairly large area of similar brick which acts as a guard ring is used. Heat is supplied to one face of this panel and the amount of heat passing through the central brick is measured by a water calorimeter surrounded by a separate water-cooled surface to make the heat flow through the guard ring similar to that through the specimen. The temperature gradient in the brick is measured by thermocouples situated on the two faces or embedded in holes a known distance apart. From the heat passing per unit area and time, and the temperature gradient, the thermal conductivity can be calculated from formula 4/1. Fuller descriptions of thermal conductivity tests are given by Norton,<sup>6.19</sup> Ezer Griffiths<sup>6.20</sup> and Clements and Vyse.<sup>6.21</sup>

6.17 HYSLOP, J. F., and BRIGGS, H. C.: "Slag and Spalling Tests on Fire-bricks," *Trans. Brit. Cer. Soc.*, **30**, p. 288 (1930–31).

6.18 A.S.T.M. *Manual of Refractory Materials*, c. 38, p. 42.

6.19 NORTON, J. H.: *Refractories* (McGraw Hill).

6.20 GRIFFITHS, E.: "Thermal Conductivity of some Industrial Materials," *J. Inst. Fuel*, **15**, 107–147 (1942).

6.21 CLEMENTS, J. F., and VYSE, J.: *Brit. Cer. Res. Assoc. Research Paper* No. 63.



**6.2.4. Resistance to Chemical Attack.** Inside a high-temperature furnace there is frequently a considerable amount of liquid slag in contact with the inner face of the refractories, whether because these refractories form a container for the liquid or because the liquid is entrained in the gas as droplets. When this liquid slag reacts chemically with the refractory, with the formation of a material of lowered melting-point, rapid disappearance of the refractories is likely to occur. Another form of chemical attack which occurs in furnaces is the carbon deposition reaction  $2\text{CO} = \text{CO}_2 + \text{C}$ : this takes place inside fireclay refractories at about  $450^\circ \text{C}$ . when a considerable amount of carbon monoxide is present and is catalysed by the presence of iron in the refractory. This process can result in the complete disintegration of the structure by the solid carbon.<sup>6.22</sup> It is liable to lead to trouble in the linings of blast furnaces, gas-producers and gas-producer mains.

Many tests have been developed for assessing liquid slag attack on refractories, but it is sufficient to describe three. In the first of these a small pellet of the slag is placed on a flat level surface of the sample and the whole is then heated for a standard time at a standard temperature. The sample is then cut through and the penetration of it by the slag is observed. Examples of this test are given in Reports 29 & 30 of the Inst. of Gas Engineers Refractory Materials Joint Committee, where 1-inch cylinders of lime were heated on the bricks at  $1400^\circ \text{C}$ . for 2 hrs. Both this test and the next are subject to the objection that in order to obtain a visible effect in a short time it is necessary to operate it at a higher temperature than will occur in practice, and this may alter the mechanism of the slag attack. This test is also subject to the objection that if the pellet becomes excessively liquid it runs away rather than penetrates the sample; all the same, it is often a valuable indicator of the relative properties of different refractories.

This latter objection is overcome in the second type of test, in which the slag sample is placed in a crucible or in a hole in the brick.<sup>6.23</sup> In this test a sample of dimensions  $80 \times 80 \times 65 \text{ mm}$ . has a hole 35 mm. deep and 44 mm. diameter drilled in it, the hole being filled with 50 grammes of finely ground slag. This test has the disadvantage that it is not the surface of the brick which is tested,

<sup>6.22</sup> HUGILL, W., ELLERTON, H., and GREEN, A. T. : *Trans. Brit. Cer. Soc.*, **32**, p. 543 (1933).

<sup>6.23</sup> D.I.N. Test Code No. 1069.



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but the surface of an artificial hole. Many bricks owe their corrosion resistance to the relative impermeability of the surface and will not show good results when tested in this way.

Larger scale tests for the action of slags on refractories aim at a closer approximation to actual conditions in a furnace. The first of these tests developed in America by Hursch<sup>6.24</sup> and by Vickers and Bell<sup>6.25</sup> in Britain, involves the spraying of slag through a gas-burner jet against the refractory. The refractory was built in the form of a vertical cylinder of 18 inches internal diameter insulated on the outside, and the gas, air and slag were introduced through a vertical axial tube bent at the bottom so as to spray the inside, this tube being rotated during the test so that the whole inner surface was sprayed; 1 lb. of slag was fed in during approximately 10 minutes. A similar test involving the direct firing of a small furnace with pulverised coal to study the effect of the coal ash on the refractory was developed by Fettke and Stewart.<sup>6.26</sup> Sufficient coal was fed to this furnace to give a total weight of ash of 500 lb. and the erosion was measured by the loss of volume of the refractory cylinder, which was arranged in a similar way to that of Hursch.

## 6.3. Available Refractories

**6.3.1. The Manufacture of Refractories.** In this section a very brief survey is given indicating how furnace design has to adapt itself to the raw materials which are supplied by nature. Much fuller discussions are given in the standard textbooks, among which can be particularly mentioned *Steelplant Refractories*, by Chesters,<sup>6.27</sup> *Refractories*, by Norton<sup>6.19</sup> and *Refractory Materials*, by A. B. Searle.<sup>6.28</sup>

Very broadly, the general process of making refractories is to take a mixture of grains of appropriate material sized from  $\frac{1}{4}$  or  $\frac{1}{8}$  inch downwards and bond this granulated material with clay in the case of fireclay or other alumina bricks, or with a fine fraction of the same material as the bricks or another compatible material, with water or other liquid to make a wet or dry paste, and mould or press the mixture to the desired shape. The size distribution

6.24 HURSCH, R. K., and GRIGSBY, C. E. : *Univ. Illinois Bull. Circ.*, 17 (1928).  
HURSCH, R. K. : *J. Amer. Ceram. Soc.*, 22, 10, p. 356 (1939).

6.25 VICKERS, A. E. J., and BELL, R. A. : *J. Soc. Glass Tech.*, 19, p. 151 (1935).

6.26 FETTKE and STEWART : *Carnegie Inst. Tech. and Mining and Met. Advisory Board. Bull.*, 73 (1936).

6.27 CHESTERS, J. H. : *Steelplant Refractories* (United Steel Companies).

6.28 SEARLE, A. B. : *Refractory Materials* (Griffin).



of the mixture of grains is of very great importance in deciding the porosity, strength and physical properties of the final brick. Coarse prefired material, called grog, is often included and may be, for example, that part of the mixture which would be retained on a 25 B.S.I. sieve. The medium size material would be that between 25 and 72 and the finest less than 72-mesh B.S.I. The percentage of very fine material (e.g. less than 1 micron), although small, is also of great importance, especially for bricks not containing much clay. The proportion of fired or hard material (grog) to clay is also an important variable.

When the shape has been moulded by hand or by machine it is dried very carefully and then fired at a controlled rate of heating to a certain maximum temperature. This firing process drives off any water of crystallisation and carbonaceous matter which may be present\* and shrinks and hardens the clay so that it will no longer soften when wetted. It also melts the more fusible mixture in the brick to a glassy matrix, according to the temperature. The greater the proportion of the matrix and the lower the porosity of the brick, the lower the resistance to thermal shock (i.e. the worse the spalling).

In the case of bricks bonded not with clay but with fine particles (preferably of the same composition as the brick itself and necessarily compatible with it) and a temporary bond, the firing makes the brick strong by a process of sintering, which is an actual growing together of the surfaces of contact of the particles, and takes place at a temperature considerably below the melting-point of these particles. The finer the particles, the lower is the sintering temperature.

From this very cursory survey it can be seen that there are four main factors governing the properties of the resulting brick, namely :

- (1) the chemical composition of the raw materials, viz. the nature of the main constituent, the amount of other materials forming impurities in the main constituent, and the composition of the bond in relation to the main materials ;
- (2) the size grading and physical conditions of the raw materials (temperature of previous firing, porosity, etc.) ;
- (3) the method of forming ; and
- (4) the firing process, particularly the maximum temperature and the duration of the firing ("soaking").

\* Many refractories, e.g. silica, do not contain water of crystallisation.



### 6.3.1 THE SCIENCE OF FLAMES AND FURNACES

In the next few subsections a summary of the properties of the more common refractories is given, these refractories being roughly classified for convenience into:

- (a) Fireclay and high-alumina bricks . . . . . (6.3.2)
- (b) Silica bricks . . . . . (6.3.3)
- (c) Basic bricks . . . . . (6.3.4)
- (d) Miscellaneous . . . . . (6.3.5)
- (e) Insulating bricks . . . . . (6.3.6)

**6.3.2. Fireclay and High Alumina Bricks.** These bricks are made mainly from natural clays and some grog from old fired

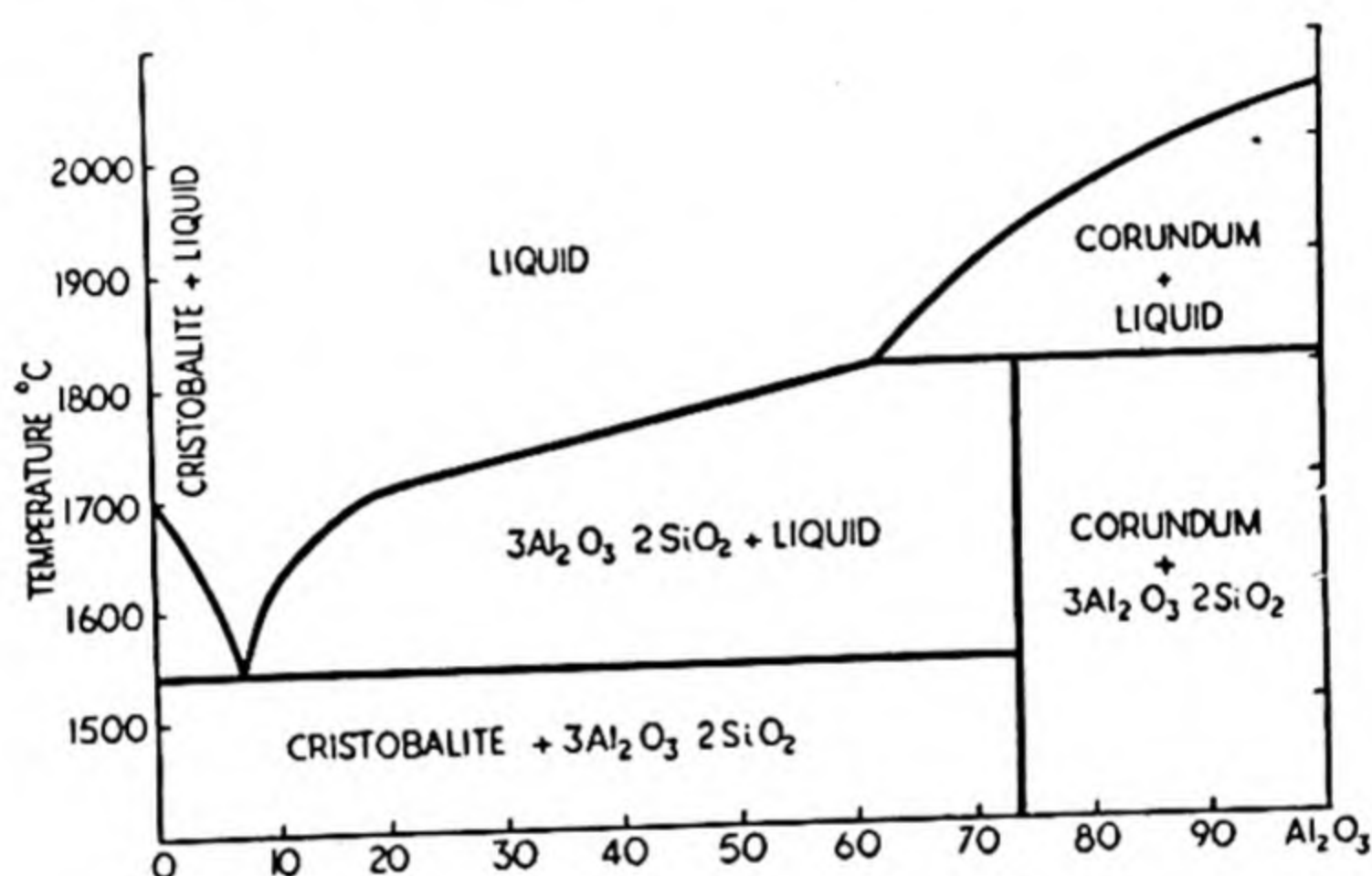


Fig. (6)2. Silica-alumina phase diagram.

bricks. The clays are formed from the weathering of feldspar-rich rocks which originally contained silica, alumina, and alkali oxides in an atomic lattice. In the course of weathering the alkalis have been more or less removed. Clays with a higher alkaline content are more plastic (ball clay), while those which are composed purely of silica and alumina (china clay) are less plastic, but considerably more refractory. Fig. (6)2 shows the silica-alumina phase diagram, from which it will be seen that pure silica melts at 1710°–1730° C., silica with 5% of alumina melts at 1530° C. and pure alumina at 2080° C.

The term “fireclay” is usually taken to refer to clays having an alumina content of 20–45%<sup>6.29</sup> with a fairly low content of iron oxide and alkaline oxides. Kaolinite (china clay) refers to the exact chemical compound  $\text{Al}_2\text{O}_3 \cdot 2\text{SiO}_2 \cdot 2\text{H}_2\text{O}$ , having alumina and

<sup>6.29</sup> ENDELL, K., FEHLING, R., and KLEY, R.: *J. Amer. Chem. Soc.*, p. 224 (1939).



silica in the proportions of 46 and 54 by weight. All bricks made from fireclay are slightly soft above  $1300^{\circ}\text{C}$ ., the actual extent depending on the firing temperature of the grog and of the brick. Those clay bricks which have high alumina and low iron-oxide contents can, however, be used up to about  $1500^{\circ}\text{C}$ ., especially if only one end of the brick is at this temperature and there is a large temperature gradient. Lower grade firebricks can only be used up to  $1200^{\circ}\text{C}$ . Fireclay brick is the cheapest refractory and is used in very large quantities in most furnaces. A great variation in the physical properties such as spalling-resistance, thermal expansion and density can be obtained by varying the grain size distribution, the clay content, and the firing temperature of the bricks: for example, for regenerator packings a brick of specially high density is obtained by high firing temperatures. The resistance of fireclay to chemical attack is only moderate, but it can be used in contact with both silica and alumina.

Next to the fireclay range of compositions, comes the high-alumina series with upwards of 45% of alumina, sillimanite\* bricks coming, so far as the chemical composition is concerned, within this range. These bricks are definitely more refractory than firebricks and have a good resistance to spalling. The under-load test curve for high-alumina bricks like that for firebrick is not sharp, but reveals slight softening at temperatures considerably below that of final failure. The latter may be as high as  $1700^{\circ}\text{C}$ ., or even  $1800^{\circ}\text{C}$ . for bricks with the highest alumina content, 5% contraction at  $1740^{\circ}\text{C}$ . being attainable with the best sillimanite bricks. Endell, Fehling and Kley<sup>6.29</sup> show that, among bricks of the alumina-silica series, those with 50% of alumina have the highest resistance to attack by slags of the coal-ash type, but this property depends also on the porosity, the densest bricks having the best resistance. High-alumina bricks are more expensive than firebricks, being made from bauxite (India and Scotland) and cyanite (India), and are accordingly used mainly for special parts of furnaces, fireboxes, arches, etc. With this type of brick, increase in density usually goes with reduction of resistance to spalling. There are many cases where high alumina bricks are not used because of the higher cost, even though the improved properties would pay in the long run.

\* Sillimanite correctly has the chemical composition  $\text{Al}_2\text{O}_3\cdot\text{SiO}_2$  with the weight percentages 63 and 37, but the name is more usually applied to a fired brick having only approximately this alumina content.



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Bricks with the highest alumina contents, ranging from 70–100%, are also manufactured, particularly those with the grog in the form of mullite, which has the formula  $3\text{Al}_2\text{O}_3 \cdot 2\text{SiO}_2$ , so that the relative weights of alumina and silica are 72% and 28%. These are very high grade bricks and are accordingly more expensive and less frequently used. They have excellent refractoriness under load, but poor resistance against coal ash attack.

**6.3.3. Silica Bricks.** Silica occurs both as a rock and as a sand of high purity and in relatively large quantities in most countries. As it has a melting-point of  $1730^\circ\text{C}$ . it is very suitable for a cheap high-temperature refractory and is widely used in open-hearth furnaces, glass tanks, coke ovens and gasworks retorts. The rock, usually ganister, is crushed and sized and the mixture rammed and bonded with a small percentage of burnt lime ("unbonded" silica bricks have, however, also been made). The resulting brick should have a silica content exceeding 96%, and when this condition is satisfied the underload curve is very sharp (see Fig. (6)1), the brick showing no sign of failure until the specimen collapses suddenly at about  $1680^\circ\text{C}$ . The point of failure is therefore within  $50^\circ$  of the melting-point of pure silica.

Silica bricks are more resistant to basic slags than their acid character would suggest. This is due to the molecular structure and to the liquid-immiscibility regions in silica systems containing such oxides as  $\text{CaO}$ ,  $\text{FeO}$ ,  $\text{MnO}$ , etc., which prevent a steep fall in the melting-point till large quantities of these basic oxides have reacted. This region is no longer found when even small quantities of alumina or alkali are present. These impurities should therefore be especially avoided. It has been stated<sup>6.30</sup> that the failure of silica roofs in open-hearth furnaces is due to fusion rather than slag attack. Silica bricks are very susceptible to spalling because silica has three physical forms which have very different densities, one of which is anisotropic. For this reason the bricks must be heated very slowly, particularly in the range  $0$ – $600^\circ\text{C}$ ., and if, for instance, a coke-oven battery built of silica is shut down, serious cracking may occur. Silica coke ovens are therefore not shut down if it can be avoided.

Silica bricks are still mainly fired in the beehive kiln which has a number of separate coal fires around the circumference and a connection to the flue in the floor under the bricks. The best bricks

<sup>6.30</sup> CHESTERS, J. H., and LEE, L: "The Properties of Magnesite and Chrome Magnesite Bricks," *Trans. Brit. Cer. Soc.*, **36** (1937).



are fired to a temperature just below  $1470^{\circ}\text{C}$ . (the temperature at which they are completely converted to cristobalite), but many are fired only to  $1300^{\circ}\text{C}$ . or  $1400^{\circ}\text{C}$ ., although they are used in open-hearth furnaces at  $1650^{\circ}\text{C}$ .

**6.3.4. Basic Bricks.**<sup>6.31, 6.32</sup> Under this heading will be included magnesite, dolomite\*, chrome-magnesite, forsterite and lime bricks. Before the war magnesite bricks were made mainly from deposits of this mineral found in Greece and Austria, and contained (after calcination) 80–90%  $\text{MgO}$ . The material was burned in a rotary kiln to drive off the  $\text{CO}_2$ , to get rid of shrinkage and reduce the tendency to hydration. It was then graded suitably for direct use, or made into bricks either by bonding with some of the magnesite ground sufficiently fine to form hydrated magnesia or by the addition of various impurities, or by the use of temporary bonds. More recently, however, magnesite has been obtained from seawater by reaction with lightly calcined dolomite, the resulting precipitate being filtered, dried and fired in a rotary kiln to  $1700^{\circ}\text{C}$ . Magnesite bricks are made from this by the above method and fired in a way similar to silica bricks. Normal commercial magnesite bricks contain only about 85%  $\text{MgO}$  and hence have a poorer under-load curve than silica bricks, because the impurities form a glassy matrix which is soft at high temperatures. Where there is little load on the bricks much higher temperatures can be employed, however, since  $\text{MgO}$  has a melting-point in the neighbourhood of  $2800^{\circ}\text{C}$ . Magnesite bricks have a very high thermal conductivity falling with rise of temperature, e.g.

at  $200^{\circ}\text{C}$ . = 26 C.H.U./ft.<sup>2</sup>  $^{\circ}\text{C}$ . hr.

$1300^{\circ}\text{C}$ . = 13 C.H.U./ft.<sup>2</sup>  $^{\circ}\text{C}$ . hr.

The bricks strongly resist attack by basic slags and coal ash, but have a poor spalling resistance.

Fairly pure dolomite ( $\text{CaCO}_3.\text{MgCO}_3$ ) occurs in large quantities in this country.

Calcined dolomite is highly refractory (the melting-point of the most fusible mixture of lime and magnesia is still  $2300^{\circ}\text{C}$ .) and has a chemical resistance at high temperatures similar to that of

\* The terms magnesite and dolomite are in general use both for the carbonates, i.e. the raw materials, and the fired bricks, i.e. the oxides.

<sup>6.31</sup> PHELPS, S. M. : "The Properties of Refractory Materials, not including Fireclays," *Tech. Bull. No. 57, Amer. Refract. Inst.*

<sup>6.32</sup> HUTTON, R. S. : "Refractories," *Tech. Publ. No. 817, Amer. Inst. Mines and Met. Eng.*



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magnesite: it would, therefore, be suitable for constructing a cheap basic refractory were it not for the fact that the oxides formed by firing rapidly hydrate when exposed to the atmosphere. In the nineteen-thirties, a new process was introduced in this country whereby the bricks made of fired dolomite were given stability by the addition of serpentine, without excessive lowering of the refractoriness. Such stabilised dolomite bricks have replaced magnesite in certain cases owing to their lower cost, but the main use of dolomite is for the monolithic hearth of the open-hearth furnace. For this purpose the dolomite mixed with coke is burned in vertical shaft kilns and the resulting "dead-burned" material is crushed to a suitable grain size distribution and rammed in the hearth of a new furnace. Erosion during each melt is made good by the process of fettling or throwing in of fresh material after the charge has been tapped.

Another basic refractory which is commonly used in the steel industry is the chrome-magnesite brick containing a mixture of chromite and magnesite. These bricks have a better refractoriness under load and resistance to spalling than magnesite bricks, but have a serious tendency to swell and flake on picking up iron oxide. The resistance of chrome-magnesite to basic slag attack is somewhat inferior to that of magnesite. The chrome ore used in the making of the brick is usually very impure and can contain several other oxides. As a result there is a tendency to form low-melting-point eutectics which may attack other refractories in the furnace. Open-hearth furnace roofs, however, have been constructed successfully with it, and it is claimed that they give a  $50^{\circ}\text{C.}$  higher operating temperature.<sup>6.33</sup> Some writers<sup>6.34, 6.35</sup> have described the special roof designs employed to take care of the greater weight and high temperature expansion of these bricks.

Chrome bricks are made from chrome ore, which occurs in Cuba, Greece, Turkey, South Africa and other places. Their refractoriness is normally above  $1800^{\circ}\text{C.}$ , but if seriously contaminated with serpentine this figure may be considerably reduced. Where the chromium oxide content is fairly high, say 40%, it is

6.33 MORAWA, F. W.: "Experiences with Special Refractories in Siemens-Martin Furnaces," *Stahl u. Eisen*, **55**, p. 201 (1935).

6.34 HUEUR, R. P., and MERWIN, F.: *Iron and Steel Engineer* (U.S.A.), **25**, No. 3, pp. 47-60 (Mar. 1948).

6.35 HUEUR, R. P., and FAY, M. H.: "The All Basic Open Hearth Furnace, European and American," *Amer. Inst. Met. Eng. Open Hearth Proceedings* 1950.



suitable for use as a refractory, with the valuable property of being *neutral*, so that, for example, it can be used to separate acid and basic bricks. Accordingly chrome bricks can be used in certain parts of the open-hearth furnace, but actually, very few are now used; the normal brick is chrome-magnesite.

Forsterite<sup>6.36, 6.37</sup> ( $2\text{MgO} \cdot \text{SiO}_2$ ) is another basic refractory which can be made from natural magnesium silicate ores at a slightly lower price than magnesite.<sup>6.38</sup> Forsterite melts at  $1910^\circ \text{C}$ . and is quite widely used in the non-ferrous metals industry in some countries.

**6.3.5. Miscellaneous Refractories.** Zircon (zirconium silicate)<sup>6.39</sup> is a slightly acid refractory which can readily be obtained with a purity of 97% from sands which occur in Florida, India and Australia. Some of the impurities in the zircon (mainly the oxides of the rare earth elements) are in solid solution. Other impurities, mainly  $\text{Fe}_2\text{O}_3$  and  $\text{Al}_2\text{O}_3$ , are concentrated on the surface of the zircon sand grains. On firing, these impurities tend to form a glass, which may account for the sintering which occurs at  $1100^\circ$ – $1800^\circ \text{C}$ . Pure zircon has a melting-point of over  $2400^\circ \text{C}$ ., and with these impurities the melting point of the commercial materials is not far below this figure. It cannot be used above about  $2000^\circ \text{C}$ ., however, as it tends to decompose to zirconia and silica, the decomposition being accelerated by certain chemicals. For refractory brick making it has the disadvantage of occurring as a fine sand (minus 72 mesh) and hence no natural grog is available.

Comstock<sup>6.40</sup> found that by making bricks with a mixture of raw and finely ground sand, then firing them at  $1600^\circ \text{C}$ . and crushing, a suitable grog could be made. Recent work has shown that this grog is not much use other than to prevent lamination when moulding, unless it is made with a mineraliser. Zircon is strongly resistant to coal ash owing to the inert, diamond-type structure of the crystal, but not very strongly resistant to basic slag or iron oxide. It has a true density of 4.6 and the bricks have a bulk density

<sup>6.36</sup> HARVEY, F. A., and BIRCH, R. E.: "Olivine and Forsterite Refractories in America," *Ind. Eng. Chem.*, **30**, p. 32 (Jan. 1938).

PLUCK, J. E.: *Iron and Coal Trades Review*, **155** (4143), pp. 257–65 (1947).

<sup>6.37</sup> GOLDSCHMIDT, V. M.: "Olivine and Forsterite Refractories in Europe," *Ind. Eng. Chem.*, **30**, p. 27 (Jan. 1938).

<sup>6.38</sup> German Patents 605371 and 650717.

<sup>6.39</sup> STOTT, V. H., and HILLIARD, A.: *J. Brit. Cer. Soc.*, Dec. 1948; *Mineralogical Mag.*, **27**, 193, p. 200 (1934).

<sup>6.40</sup> COMSTOCK, G. F.: *Amer. Cer. Soc.*, p. 12 (1933).



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of 3.5, so that they are nearly twice the weight of silica bricks of equal size. Zircon has not yet found wide industrial application.

Silicon carbide ( $\text{SiC}$ ) is made from carbon and sand in electric furnaces.<sup>6.41, 6.42</sup> The material is packed together between two conductors and a current is passed through it, heating the central core to a very high temperature. The process is only economic where water power enables electricity to be obtained cheaply, and most of the world's silicon carbide comes from Niagara Falls. The central core, which has been heated to the highest temperature by the current, is used by the abrasive industry, and the layers immediately surrounding it are used for refractories. They are usually bonded with clay, with the result that although silicon carbide itself has a melting-point in the neighbourhood of  $2400^{\circ}\text{C}$ . and commercial silicon carbide refractories have a refractoriness under-load better than  $1800^{\circ}\text{C}$ ., the clay tends to run out at temperatures above  $1500^{\circ}\text{C}$ . The clay bond has also a very poor resistance against chemical attack. Silicon carbide refractories have a high thermal conductivity and a tendency to oxidise slowly in the temperature range  $900^{\circ}\text{C}$ – $1300^{\circ}\text{C}$ . in air, although they can be used for a good time at  $1600^{\circ}\text{C}$ . When the bricks oxidise, they swell, because the  $\text{SiO}_2$  molecule is larger than the  $\text{SiC}$  molecule. They have a very good resistance to coal-ash attack and adhesion, and are thus frequently used in water-gas generators or chain-grate stokers at places where the coal ash tends to collect and adhere to the refractory.<sup>6.43</sup> Where the coal ash is high in iron oxide there is a tendency for the silicon carbide to be destroyed by the formation of iron silicate. Silicon carbide is also sometimes used for the conduction parts of recuperators.

Carbon bricks<sup>6.44</sup> are made from coke or anthracite bonded with pitch and fired at  $1100^{\circ}\text{C}$ . to give a practically pure carbon block with high thermal and electrical conductivities. This material naturally oxidises in  $\text{CO}_2$  or  $\text{O}_2$  and can only be used in strongly reducing atmospheres. It has, however, a very good chemical resistance to slags, a good resistance to spalling and a very high

6.41 SINGER, F.: "Ofen" (Handbuch der Chemisch-technischen Apparate, A. J. Keiser, p. 1203).

6.42 RUFF, O.: *Trans. Electrochem. Soc.*, **68**, pp. 87–108 (1935).

6.43 GEIGER, C. F., TURNER, A. A., and STACH, O. R.: *Chem. Eng. Prog.*, **144**, pp. 933–936, C.4v(5), C.4e(11) (Dec. 1948).

6.44 RIGBY, G. K., BOOTH, H., and GREEN, A. T.: "An Investigation on the Manufacture of Carbon Blocks for Blast Furnaces," *Trans. Brit. Ceram. Soc.*, **41**, 144 (1942).



refractoriness-under-load. In America and Germany, carbon blocks have been used for lining some blast furnace hearths, whilst in England this development has been carried further, resulting in the all-carbon blast furnace lining.<sup>6.45</sup> Carbon in the form of graphite, obtained in Ceylon or Madagascar, is also used (bonded with clay) for the manufacture of metal-melting crucibles. In this case, as in that of silicon carbide, the refractoriness is largely decided by the clay bond.

**6.3.6. Insulating Bricks.** Two main types of insulating bricks are used in the construction of furnaces. The first of these consists of bricks made from diatomaceous earth (diatomite). This consists of the silicious skeletons of small organisms and is found in large quantities, for example, in Ireland, Denmark and N. Germany. It has a very porous structure and hence is rendered a good insulator by the number of stagnant pockets of air. When made into bricks it can be used up to  $900^{\circ}\text{C}$ . Such bricks are extremely light ( $20\text{--}40\text{ lb./ft.}^3$ ) and also soft and weak. They have a high permeability to gas, passing about ten times as much air through as ordinary firebrick.<sup>6.46</sup> They can be readily cut and drilled, a property which is often of use in experimental work. Diatomite insulating bricks must be used very carefully.

The second type of insulating brick is the porous firebrick, which consists of a clay brick made in the same way as a normal firebrick except that a proportion of sawdust is added to the mixture. On firing, this sawdust burns out and leaves a porous structure in the brick which gives it insulating properties by virtue of the air cells. Such bricks when made from good-quality clay can be used with a hot-face temperature of  $1400^{\circ}\text{C}$ . and special bricks with high alumina contents have been made which can be used to  $1600^{\circ}\text{C}$ .\*

Porous firebricks are soft and can be drilled and cut, but they have better mechanical strength than the diatomaceous earth bricks. In recent experiments graded bricks varying from dense firebrick at one side to a material of high porosity at the other, have been made. Such bricks have the obvious advantage that

\* B.P.562,788 describes the manufacture of thick-walled bubbles from a fused material consisting of  $\text{Al}_2\text{O}_3$  with about 1% of  $\text{TiO}_2$ . These are bonded with pure kaolin to form a brick which is strong and highly refractory as well as having good insulating properties.

<sup>6.45</sup> CHESTERS, J. H., ELLIOT, C. D. and MACKENZIE, J.: "Evolution of the All-Carbon Blast Furnace," *J. Iron and Steel Inst.*, **167**, Pt. 3, p. 273 (1951).

<sup>6.46</sup> LEUN, A. V.: "Insulating Refractory Brick: Their Properties and Application," *Trans. Amer. Foundrymen's Assoc.*, **45**, p. 274 (1937).



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they combine a tough and refractory inner wall with an insulating outer surface, but they are not yet on the market owing to the difficulty of manufacture.

It is possible to build furnaces completely from porous firebricks, giving the inner face a coating with a refractory cement to reduce gas permeability. Such a furnace is not only well insulated but has a very low thermal capacity, so that the amount of heat needed to raise its temperature is kept to a minimum. This can result in a very great fuel saving in processes where it is necessary to insert a charge into a cold furnace and heat them up together. Porous bricks cannot, of course, be used where walls are liable to be attacked by slag or abraded by the charge, and it is therefore rarely practicable to use furnaces entirely made from them; their main use is therefore for insulation of structures made from hard bricks.

#### 6.3.7. Tabular Summary of the Properties of Refractories.

Table 6.1 is a general summary of the preceding five subsections in which the four main properties of refractories set out in 6.3.1 are very roughly stated for each type. It should, however, be clearly noted that there is an enormous variation in the properties of bricks with the same name, and that such a table can only be an approximate guide to the designer in choosing the most suitable bricks.

#### 6.3.8. The Use of Pure Oxides as Refractories.

It would appear to be clear on scientific grounds that a refractory should not be used at a temperature higher than that at which it has been fired. It is, however, a fact that in nearly all high-temperature industrial processes the firing temperature of the refractories is, owing to cost considerations, well below that of use, and the furnace made of these refractories has to be fired especially carefully because of these limitations. Since low firing temperatures are economical, fluxes (natural impurities or additions) must be present to produce a hard brick. When super-refractories are required, and the cost is justified, pure materials and therefore high firing temperatures can be used.

Pure oxides provide very valuable refractories and a number of important papers describe their use,<sup>6.47, 6.48, 6.49 and 6.50</sup> this use

<sup>6.47</sup> STOTT, V. H.: "The Manufacture of Refractory Articles from Pure Oxides of High Melting Point," *J. Cer. Soc.*, **48** (1949).

<sup>6.48</sup> RYSCHKEWITZ, E.: "Refractories for Work at High Temperatures," *Chem. & Met. Eng.* (Feb. 1932).

<sup>6.49</sup> TURNER, D.: "Special Refractories for Metallurgical Research at High Temperatures," *Trans. Eng. Cer. Soc.*, **33**, p. 33 (1933-34).

<sup>6.50</sup> PIRANI, M.: *Elektrothermie*, p. 215 (Berlin, 1930).  
SINGER, F.: "Gas-tight Sintered Alumina Ware," *Trans. Ceramic Soc.*, **37**, p. 346 (1938).



TABLE 6.1 GENERAL SUMMARY OF PRINCIPAL REFRACTORIES

Refractory brick	Density		Thermal conductivity C.H.U. in./ ft. <sup>2</sup> °C. hr.	Mean heat capacity at 600° C.		P.C.E., °C.	Refractoriness underload, °C.	Chemical resistance	Spalling resistance	Coefficient of thermal expansion 20°–1400° C.
	lb./ft. <sup>3</sup>	gm./c.c.		cal./gm. °C. C.H.U./lb. °C.	C.H.U./ft. <sup>3</sup> °C.					
Fireclay	130	2.1	8–14†	0.23	30	1500°–1700°	Gradual failure 1300°–1500° 1650°–1750° 1750° Sharp fail 1680°–1700°	Moderate up to 1400° C. Good—neutral " Strongly acid. Good against iron oxide Good against iron oxide and basic slag	Good " Poor at low tem- peratures Poor	— 5.9 10–6 5.4 9.0
Sillimanite Alumina 80% Al <sub>2</sub> O <sub>3</sub> Silica	135 165 112	2.2 2.6 1.8	8–10 8–10 8–14	0.17 — 0.23	23 — 26	1800° 1900° 1710°	Usually lower than silica	"	"	13.2
Magnesite	150	2.4	14–19	0.27	40	2000°	"	Bad Poor unless very pure	Moderate "	10.3 8.9
Chrome-magnesite Chromite	171 203	2.7 3.3	13 10–13	— 0.22	— 45	1700°–1900° —	Depends on purity and firing tem- perature	Acid; good against coal ash	Good	4.5
Zircon	200	3.2	12–22	—	—	2000°	1680°	Good except against oxi- dation	"	—
Carborundum	170	2.7	20–50	0.28	48	2000°	1800°–1900°	"	"	—
Carbon (clay-free)	97	1.6	70	0.40	39	2000°	—	Low	"	—
Insulating firebrick	{ 50 70	{ 0.80 1.12	2.0 3.5	0.23 0.23	{ 12 14	Used up to 1400°	—	"	"	—
Diatomaceous earth	{ 40 30	{ 0.64 0.48	1.5 1.0	0.23 0.23	{ 9.2 6.9	Used up to 900°	—	"	"	—

\* Strictly speaking the P.C.E. (pyrometric cone equivalent) refers to the bending at a particular temperature but under standard conditions the equivalent temperatures given may be used.

† The thermal conductivity of aluminous firebrick varies with temperature approximately as follows:

Temperature °C.		300	600	900	1300
Thermal conductivity (cal./cm.-C.-sec.)		0.0022	0.0027	0.0031	0.0036
Thermal conductivity (C.H.U. in./ft. <sup>2</sup> °C. hr.)		6.4	7.7	8.9	10.5

(Figures supplied by Brit. Ceram. Research Association).



### 6.3.8 THE SCIENCE OF FLAMES AND FURNACES

being, however, limited to the laboratory scale. The following table shows the melting points of various oxides:

TABLE 6.2

PURE OXIDES SUITABLE FOR USE AS HIGH-GRADE REFRACTORIES

Oxide.	Approximate melting-point, ° C.
Al <sub>2</sub> O <sub>3</sub>	2080°
MgO	2800°
BeO	2500°
TiO <sub>2</sub>	1825°
ZrO <sub>2</sub>	2800°
Spinel, MgO.Al <sub>2</sub> O <sub>3</sub>	2000°
Zircon, ZrO <sub>2</sub> .SiO <sub>2</sub>	2400°-2500°
ThO <sub>2</sub>	3000°

Shapes can be made from the oxides without the addition of impurities by two main methods. The first of these is slip casting in which very finely ground material is used to make a fluid slip and this is cast in a plaster-of-Paris mould. The grinding needs to be sufficient to make more than 20% of the material less than  $1\mu(10^{-4} \text{ cm.})$  in size. The second method consists of moulding materials of suitable grain size distribution under pressure with a temporary bond such as starch, a fair proportion of fine material being necessary. The shapes made by either of these methods can then be fired to temperatures of the order of 1800° C., and result in materials of a refractoriness far exceeding that of ordinary bricks.

Hutton<sup>6.51</sup> has pointed out the possibility of preparing such oxides on a large commercial scale. In his opinion, high production costs would be largely overcome once the processes were going on on a large scale; the purification of natural raw materials has already been achieved in other industries, the cost dropping tenfold or more as the process develops. The cost of firing to 1800° C. could be greatly reduced given a kiln refractory which, even when thoroughly insulated and allowed to reach about 1800° C. throughout, would not soften.\* Since the firing does not involve endothermic reactions the counterflow principle could be fully developed and the temperature attained without excessive fuel consumption. It is doubtful whether the bulk production of super-refractories is being developed. It is quite certain that the maximum temperatures

\* It is not certain, however, that any material remains perfectly unplastic and non-sticky above 2000° C.

<sup>6.51</sup> HUTTON, R. S.: "Refractories," *Amer. Inst. Mining and Met. Eng.*, No. 817 (June 1937).



obtainable in industrial furnaces will remain seriously restricted until such refractories become available.

## 6.4. The Principles of Wall and Crown Construction

**6.4.1. Walls.** When a furnace structure is heated it cannot expand uniformly as a unit because of the temperature gradient through the walls. Hence walls must be made up from comparatively small units in order to replace irregular cracking on heating by a regular set of joints. It follows from this that it is necessary in building the furnace, either to leave gaps for expansion at the joints or to allow the wall as a whole to expand, in which case the joints simply allow differential expansion to take place. There are, therefore, a whole set of practical methods for leaving gaps in the brickwork for expansion, either open or filled with a material which burns out. Similarly, many good ways of laying the bricks with interlocking structure to give the whole wall stability are current in the *art* of furnace bricklaying. They do not, however, concern the principles of furnace *science*, since they do not restrict the structures which are possible, although they have the very greatest practical significance. For this reason they are not discussed here ; for a discussion of these methods consult Norton<sup>6.52</sup> and Buell.<sup>6.53</sup>

The statics of furnace walls is quite straightforward : the load on the bricks at any point is calculated directly from the weight of the bricks above it. The general principle of furnace construction where the refractories are insulated and used to the limit of their temperature should be to hang as much as possible of the weight of the wall and of the crown on the outer steel structure by means of brackets. The limitation of this process is, of course, that if metal brackets are used to support the heavier inside bricks, they may overheat (if they actually melt the effect on the refractories is liable to be disastrous). To avoid this the heat losses must be increased at the hot end of the brackets; e.g. it is possible to have a gap in the insulation so that the bracket is cooled by natural convection, or to have water cooling of brackets completely immersed in the brickwork; this implies, however, a considerable departure from the ideal insulating brick wall. In addition, the leakage of hot gases out through the furnace wall near the metalwork must be especially avoided. Just as the strength of the furnace should be supplied by external metalwork, so the general gas-tightness of the

<sup>6.52</sup> NORTON, F. H. : *Refractories*, Chap. 24.

<sup>6.53</sup> BUELL, W. C. : *Open Hearth*, 2.



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furnace should be attained, as far as possible, by the use of a steel-plated outer structure, since the joints between the bricks always open up more or less in practice.

**6.4.2. Furnace Roofs.** There are two main ways of covering the top of a hot chamber. One of these is the masonry *arch* of wedge-shaped bricks which spans the furnace and is held up by an inclined pinching force from the sides. This force is supplied by bricks, known as skewbacks, which are supported in a steel channel along the top of the side walls and held together by vertical steel girders using horizontal tie rods. The other method of spanning the furnace roof is the suspended arch, which is discussed in 6.4.5. As mentioned in 6.4.1 for the case of walls, ideally the skewbacks

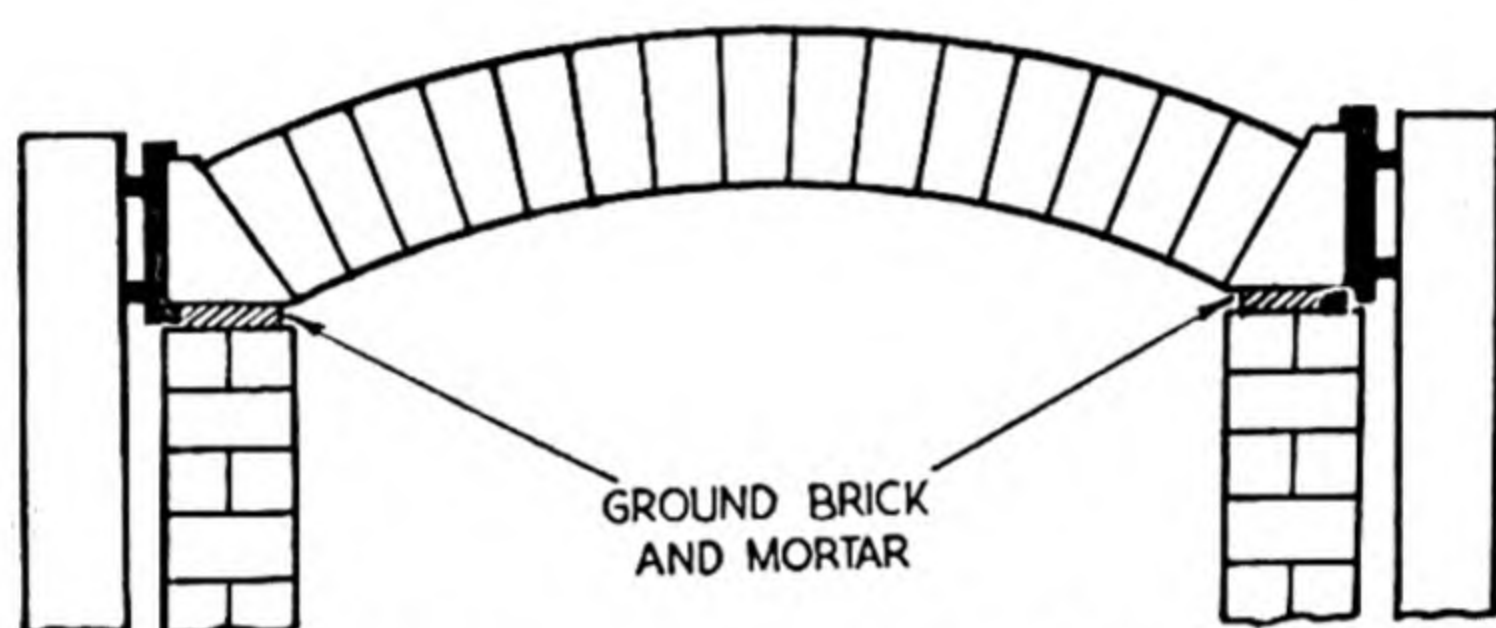


Fig. (6)3. Separate Furnace Roof Support.

and hence the crown should be supported entirely from the steel frame. A typical arrangement by which this can be done is shown in Fig. (6)3.

The roof can be flat or curved. When it is flat (Jack arch) each brick has a special shape, and such arches are only used in particular cases. The curved arch is more stable and can be used to span much greater distances, but it does dictate the shape of the furnace inside, since the centre has to be appreciably higher than the edges.

The statics of the circular arch are considered next, since it is in such arches that there is a possibility of bricks receiving a load comparable with that of the under-load test discussed in 6.2.2.

To evaluate the forces in an arch such as that shown in Fig. (6)3 let  $r$  be the mean radius of the arch (ft.) ;

$2\theta$  the angle subtended by it at the centre (in radians) ;

$W$  the total weight of the arch (lb./ft.-run) ;



$H$  the horizontal force exerted on it by the skewbacks ; and

$R$  the resultant of  $H$  and  $W/2$ , i.e. the total force exerted on the end of the arch by the skewbacks.

It is assumed that  $R$  must act normally to the face of the end brick,  $s$  is the span of the arch in ft.,  $d$  its thickness in inches and  $\rho$  its density (lb./ft.<sup>3</sup>) (see Table 6.1).

From the geometry it follows that

$$s = 2r \sin \theta \text{ ft.} \quad \dots \dots \dots (6/1)$$

and the rise of centre of arch  $= r(1 - \cos \theta)$  ft.

$$\begin{aligned} W &= 2\rho \cdot r \cdot \frac{d}{12} \cdot \theta \\ &= \frac{\rho \cdot \theta \cdot s \cdot d}{12 \cdot \sin \theta} \quad \dots \dots \dots (\text{from } 6/1) \end{aligned}$$

From the triangle of forces at one end, it follows that

$$R = \frac{W}{2} \operatorname{cosec} \theta$$

and

$$H = \frac{W}{2} \cot \theta$$

From these equations, the following table is obtained, by a similar method to Table 10, p. 715, of Norton's *Refractories* (see 6.33).

TABLE 6.3  
STATICS OF CIRCULAR ARCH

Rise in inches Span in feet	$2\theta^\circ$	$\frac{\theta}{\sin \theta}$	Horizontal thrust = $\frac{H}{W}$ Weight	Normal thrust = $\frac{R}{W}$ Weight
1.0	38	1.02	1.45	1.53
1.5	56	1.04	0.93	1.06
2.0	74	1.07	0.65	0.82
2.5	90½	1.12	0.50	0.72
3.0	106	1.16	0.39	0.62

To calculate the forces for the weight of an arch of known rise and span from this table, the third column is multiplied by  $\frac{\rho \cdot s \cdot d}{12}$  to give the weight  $W$  and from this the horizontal and normal thrusts can be calculated from the fourth and fifth columns. It can be seen from this table how, for a rise greater than 1.61 in./ft., the normal thrust of the arch is less than the weight to which the lowest bricks would be subject if the whole arch were placed on edge. Thus, for



## 6.4.2 THE SCIENCE OF FLAMES AND FURNACES

example, in a 13 ft.-span arch of silica brick of density 130 lb./ft.<sup>3</sup> the normal thrust per in.<sup>2</sup> at the end would be 8.5 lb. with a 3 in./ft. rise and 18 lb. with a 1 in./ft. rise. Since arches are rarely built with a span greater than 13 ft. it will be seen that in silica bricks the under-load test is more severer as far as load is concerned than actual conditions. It should be noted, however, that a brick of twice the density will be subjected to twice the thrust, so that heavier bricks are more likely to reach the test figures of 28 and 50 lb./inch.<sup>2</sup>

In cases where the arch requires covering with further brickwork, or insulating material, it is often desirable to spring this arch independently.

**6.4.3. The Catenary Arch.** For rises above a certain amount, the circular arch can only be built because of frictional forces between the bricks. It has been shown, however,<sup>6.54</sup> that an arch built in the form of an inverted catenary can be stable with any ratio of rise to span. If a piece of heavy string is held apart at its two ends at a distance less than its length, the shape it takes up is a catenary, and the suspending force is equal to the tension everywhere along it. Thus the inverted catenary is the natural shape for a heavy arch where the compression across every brick face is normal. Norton has shown that since the weight increases with the rise for a constant span, and the factor of thrust over weight decreases with the rise, the thrust is a minimum when the rise divided by the span is 0.56, i.e. a rise of 6.7 in./ft. Such an arch can be readily drawn for a given span and rise by marking out the ends and centre point on a sheet of drawing-paper, and hanging a chain with small links so as to pass through all three. The weight of the crown can be calculated by measuring the length of the curve and the horizontal and normal thrusts are given by:

$$\frac{W}{2} \cot \theta \text{ and } \frac{W}{2} \operatorname{cosec} \theta \text{ respectively} \quad . \quad . \quad (6/2)$$

where  $\theta$  is the angle between the tangent at the ends of the catenary curves and the horizontal. This shape of roof is more suitable for carrying insulation than the circular shape. The great advantage, however, is that the side walls and roof form one continuous catenary arch.

Norton recommends constructing the catenary arch by means of a mixture of two shapes of brick, having respectively the tapers corresponding to the ends and the centre, but this will, of course,

<sup>6.54</sup> NORTON: *J. Amer. Cer. Soc.*, p. 144 (Mar. 1926).



cause a slight deviation from the exact condition of normal stresses at each joint.

**6.4.4. Expansion Stresses in Arches.** When the furnace system is heated the arch is subjected to additional stresses for two reasons. Firstly, the expansion of the whole crown causes it to rise and squeeze the hot face of the arch, an effect which is less for high arches than for low ones. The ideal solution to this type of additional stress is to allow the skewbacks to move apart in proportion to the expansion, so that the crown does not rise at all but in fact flattens. This is done by slackening the nuts on the tie-rods as the furnace heats. In some furnace designs springs on the tie-rods have been introduced to keep the compression more or less constant.

The second kind of heating stress is that which arises due to the temperature gradient through the brick causing greater expansion on the inner face than on the outer. This type of thermal stress is very much reduced by insulating the bricks and so reducing the temperature gradient. Recently a method of measuring these stresses in an actual open-hearth furnace roof has been developed by building two water-cooled plates into the arch, the thrust between which is taken on gauges situated near the top and the bottom respectively.

**6.4.5. Suspended Roofs.** Many suspended roof designs have been worked out in which bricks have been hung individually from steel bars, either by means of tie-rods hooked into the bricks or by clamps fitted over projections in the bricks, or by water-cooled tubes through the bricks. In some designs several bricks are supported from a single hanger, by hooking one over another, and a further modification is to have interlocking surfaces on the bricks so that broken brick pieces cannot fall into the furnace. A number of typical constructions are shown on p. 751 of Norton's book. Suspended bricks can span a furnace of any desired width and are particularly convenient for constructing overhanging corners and special roof contours. Thus suspension allows greater freedom in the choice of the shape of the furnace interior; in particular, it makes it unnecessary to have arch-shaped roofs. The main disadvantage of suspended roofs is that if insulation is placed around the metal hangers they will overheat, while if these hangers are water-cooled, insulation is not very effective. In the type of roof in which the bricks are suspended by projecting pieces which are clamped it is possible to lay insulation between the projections and



## 6.4.5 THE SCIENCE OF FLAMES AND FURNACES

thus insulate about half the area.\* Suspended roofs are also suitable for furnaces made entirely of high-temperature insulating brick.

ALL JOINTS BETWEEN AIR AND  
WASTE GAS FLUES ARE HORIZONTAL.  
WEIGHT OF TILE KEEPS JOINTS TIGHT.

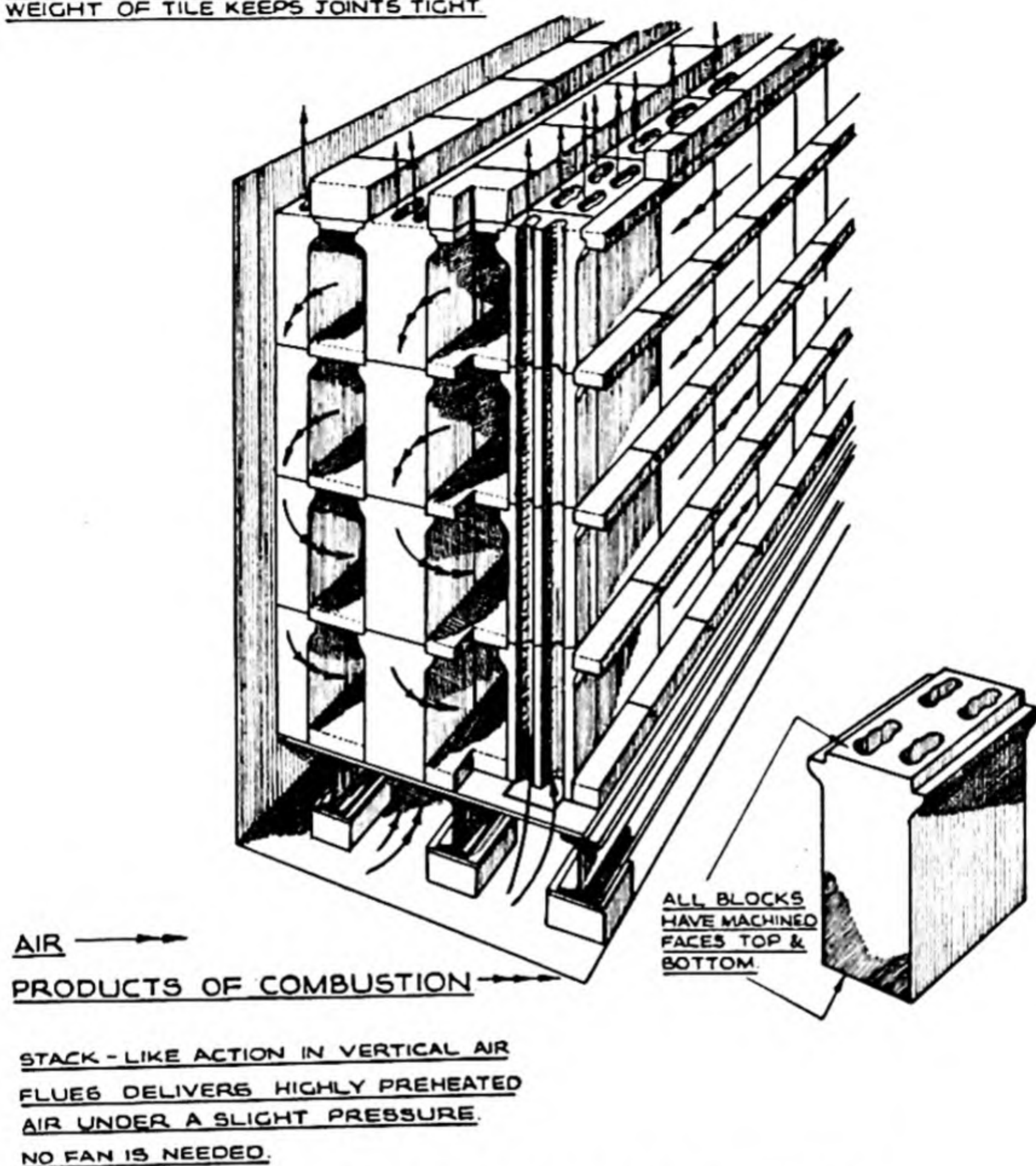


Fig. (6)4a. The Stein Refractory Recuperator.  
(Courtesy of Stein & Atkinson Ltd.)

## 6.5. Internal Walls

**6.5.1. Basic Principles.** Unlike external walls, internal partitions cannot depend either for their strength or their gas-tightness upon metal reinforcing. They also differ from the outer skin in that for inner walls high conductivity is usually desirable and there is certainly no necessity for low conductivity. Examples of inner walls

\* A recent patent (B.P. 563,593) enables a higher percentage of the roof to be insulated by using refractory hangers of relatively small area which are independent of the main bricks. The bricks of the roof hang from these by specially shaped T-slots.



are the gas ports in an open-hearth steel furnace, muffles (the effect of these on heat transfer has been discussed in 4.3.4, where it was seen that in many cases the thermal resistance of the gas film on

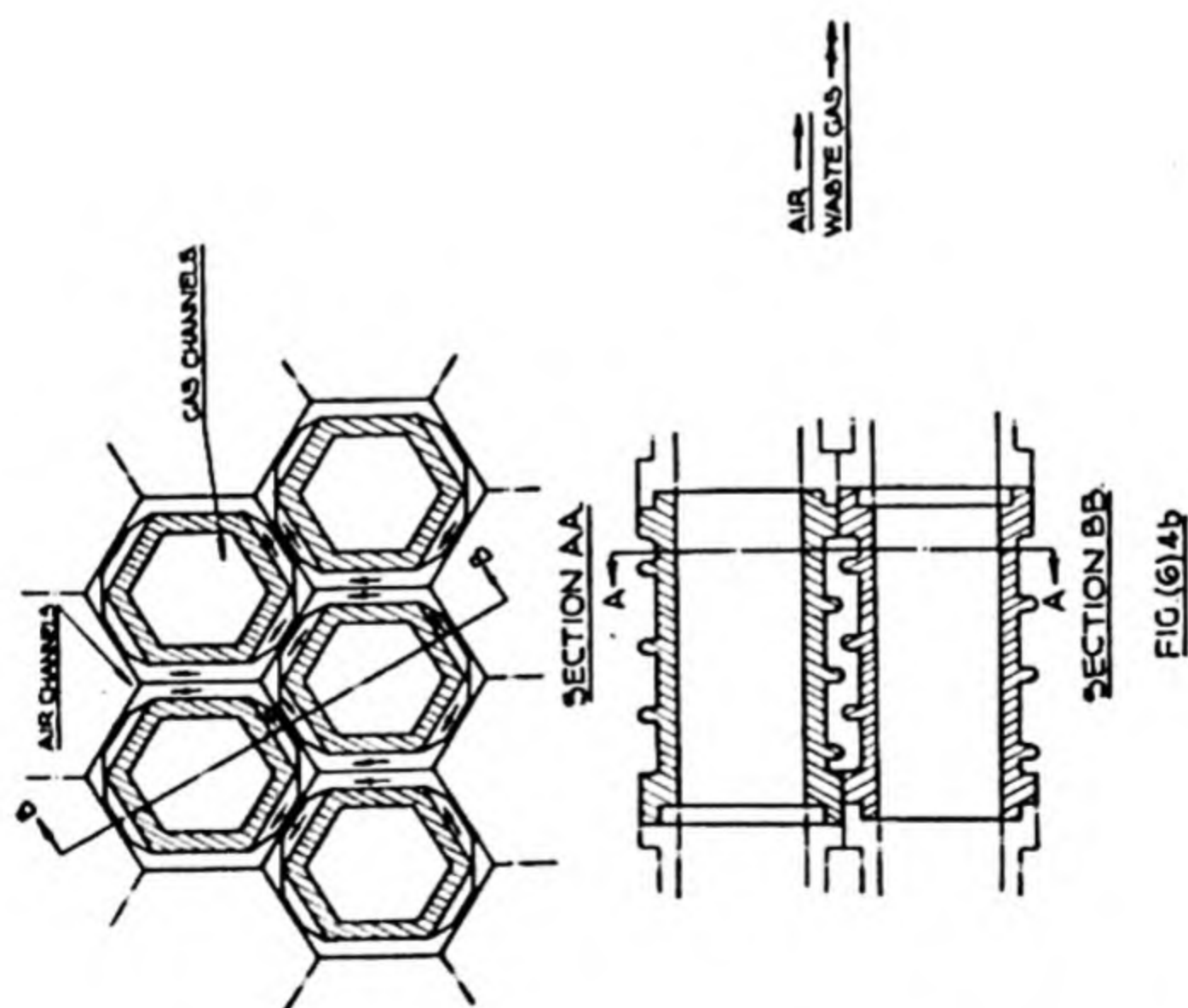
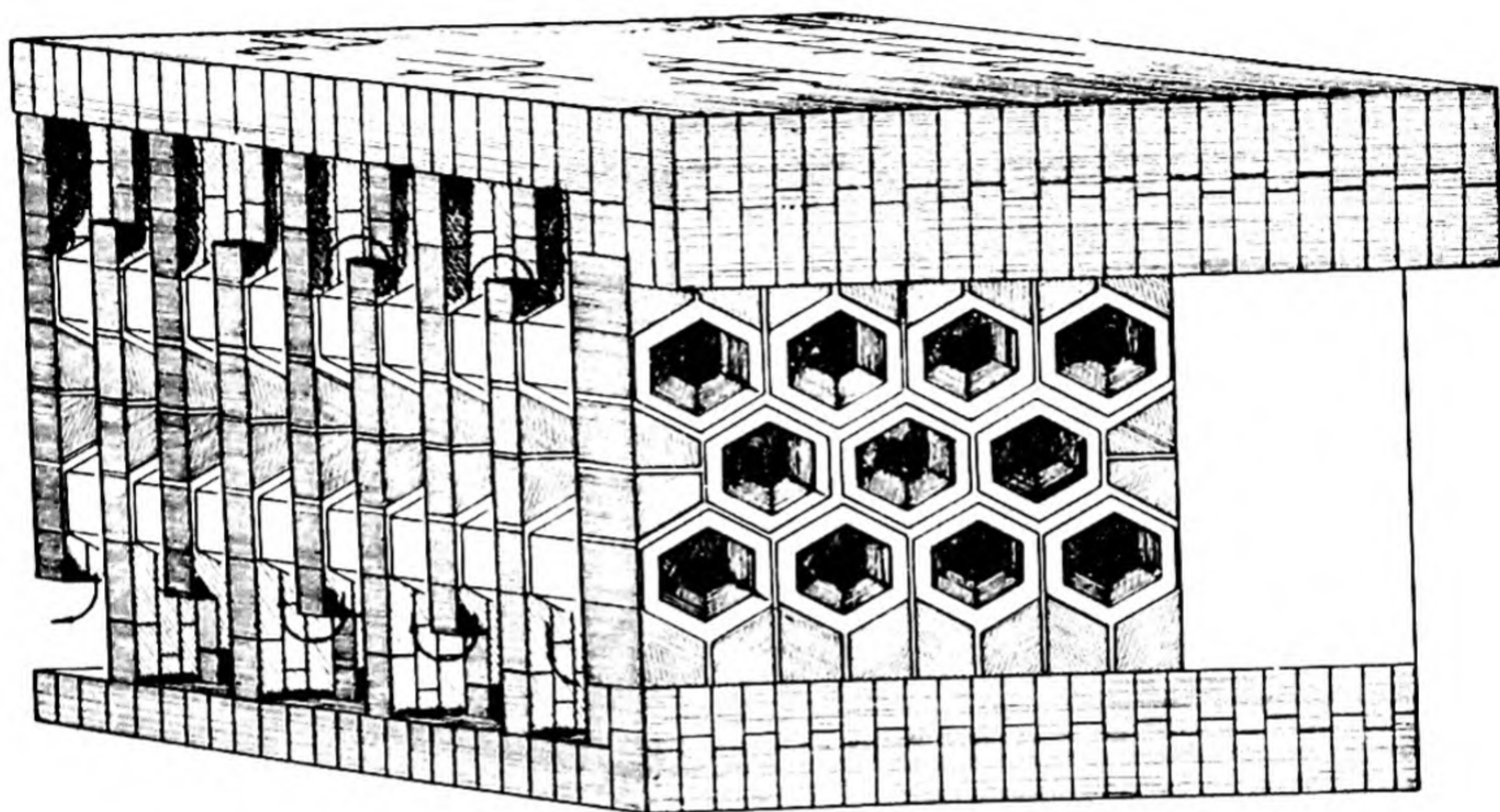


Fig. (6)4b. The Teisen Hexagonal Refractory Recuperator.

(Courtesy of Th. Teisen, C. E.)

the surface of the muffle exceeds that of the muffle), heat exchange tubes in gas heaters such as recuperators, the walls of coke ovens and the retorts of gasworks furnaces.



## 6.5.1 THE SCIENCE OF FLAMES AND FURNACES

The basic scientific principles for the construction of internal walls are :

- (1) To obtain gas-tightness by using small pieces with labyrinth seals, e.g. tongues and grooves or other interlocking arrangements. Owing to expansion, such grooves will never be tightly closed, but can be made to have high resistance to gas leakage. An example is given in Fig. (6)4 of two types of recuperator in which the gases have to pass two seals before mixing.
- (2) High conductivity in walls is obtained by using dense bricks such as firebrick of very low porosity or else materials of very high thermal conductivity such as carbon or carborundum.
- (3) Strength of internal partitions has to be obtained by very careful design. There are very many ways of doing this, but the general principle is to have no long span of unsupported refractory, since nearly all of them become more plastic the higher the temperature.

## SUMMARY OF CHAPTER 6 THE SCIENCE OF FURNACE CONSTRUCTION

### 6.1. The Furnace Skin

Furnace design cannot fulfil perfectly the requirements stated in the previous chapters owing to limitation of the available materials: the science of construction is concerned with fulfilling them as nearly as possible in spite of these limitations.

### 6.2. Tests for Refractories

**6.2.1. The Purpose of Refractory Tests.** Laboratory tests are intended to indicate how well a refractory will behave in a furnace, but they can never do this perfectly.

**6.2.2. Dimensional Stability under Working Conditions.** Usually, refractoriness is expressed in terms of the pyrometric cone equivalent which is given by that standard cone which just bends to the base at the same time as a cone of the refractory. The cone number can be roughly related to a temperature value. The refractoriness-under-load test gives a measure of the high temperature strength. There are a number of "standard" tests for permanent expansion or contraction. A cheap, moderately reliable



test for spalling characteristics is that in which a single brick or specimen is heated and cooled until it fails. A larger and more simulative test employs a panel of bricks.

**6.2.3. Thermal Conductivity.** At low temperatures thermal conductivity can be measured accurately in the laboratory, and provides useful data for the designer.

**6.2.4. Resistance to Chemical Attack.** Accelerated tests using pills and crucibles require a higher temperature than the operating temperature of the brick in service and may therefore introduce errors; spraying the refractory with liquid slag is more reliable but expensive.

### 6.3. Available Refractories

**6.3.1. The Manufacture of Refractories.** The properties of a brick depend mainly on the chemical composition, physical state of the raw materials, and the subsequent processing.

**6.3.2. Fireclay and High-alumina Bricks.** These are made from clays with the addition of high-alumina bauxite. They have a wide softening range and only the high-alumina ones are useful above 1500° C.

**6.3.3. Silica Bricks.** Silica bricks retain their strength up to 1680° C. and among other applications are suitable for open-hearth steel furnace crowns, but they require very careful heating.

**6.3.4. Basic Bricks.** Magnesite bricks are very resistant to coal ash and basic slag, and if pure can be used at higher temperatures than silica. The most popular bricks are made of chrome-magnesite.

**6.3.5. Miscellaneous Refractories.** Chrome bricks are said to be chemically "neutral," but are rarely pure enough to have good physical properties at high temperatures. Other special materials are zircon, silicon carbide and carbon.

**6.3.6. Insulating Bricks.** The most common insulating bricks are made from diatomaceous earth (hot face up to 900° C.) and porous firebrick (hot face 1400° C.).

**6.3.7. Tabular Summary of the Properties of Some Refractories.** Table 6.1.

**6.3.8. The Use of Pure Oxides as Refractories.** Some pure oxides might provide excellent bulk refractories for certain purposes if they could be prepared and fired to high temperatures commercially.



**6.4. The Principles of Wall and Crown Construction**

**6.4.1. Walls.** The general principle should be to hang as much as possible of the weight of the brickwork directly on the steel frame.

**6.4.2. Furnace Roofs.** Table 6.3 gives figures for calculating the thrust of an arch when the weight and rise/span are known.

**6.4.3. The Catenary Arch.** When it is necessary to build an arch with a rise greater than 3 inches per foot span, the catenary shape is more stable than the circular. This can be designed using a chain and built with bricks of two tapers.

**6.4.4. Expansion Stresses in Arches.** Heating the furnace causes additional arch stresses due to overall expansion and differential expansion between the hot and cold faces; the first can be reduced by adjusting the tie-rods, the second by insulation.

**6.4.5. Suspended Roofs.** Suspended roofs give more freedom of design but are difficult to insulate.

**6.5. Internal Walls**

**6.5.1. Basic Principles.** Internal walls are made fairly gas-tight by having interlocking sides to the bricks : they must be very well supported since the bricks are heated right through.



## 7. THE APPLICATION OF THE SCIENTIFIC METHOD TO FURNACES

### 7.1. The Limitation of the Scientific Method

Since a large part of the aim of the present monograph is to facilitate the application of scientific results and the scientific method to industrial furnace design and practice it is appropriate to end with a chapter which attempts to survey the extent to which one can hope to apply the scientific method. The scientific method is essentially concerned with the study of *exactly specified, isolated systems*. The laws of science can only be formulated, and theories to explain these laws can only be constructed, for such systems. For example, in a specific problem concerned with the heat transfer from a fluid to a solid surface the quantitative law connecting the heat transfer with the significant variables such as the temperature difference and size can be given provided that the geometry of the system and the nature of the fluid and the surface are all similar to those on which experimental work has previously been carried out. This illustrates how the system must be exactly specified ; that it must be isolated is shown by the fact that the actual heat transfer, for example from the surface to the fluid, will not be that calculated from the formula if there are other processes such as a chemical reaction or an electric current also affecting the heat distribution in the system.

It is of course true that these conditions are never perfectly satisfied, so that the theoretical system with which science can deal is never perfectly realisable. In the laboratory, however, the errors due to deviations from these conditions can usually be reduced to insignificant proportions by suitable design of the apparatus, coupled with sufficient care in carrying out the experiment. For example, in measuring the thermal conductivity of a slab of material care can be taken that there is no source of heat other than the measured one, the specimen can be surrounded by a "guard ring" to avoid heat flow in a direction other than the required one, and it can be cut accurately to a simple geometrical shape such as a cylinder with parallel faces, so that the relation of the measured heat transfer to the thermal conductivity is given accurately by the simple formula.



## 7.1 THE SCIENCE OF FLAMES AND FURNACES

In industrial practice, on the other hand, the deviations of a system from the simple ones with which a theory can deal are usually very large and hence it is necessary to make careful allowance for these deviations in applying the results of science. Theoretical treatment is by its very nature unable to make this allowance, which is essentially a matter of practical experience. For this reason a great deal of furnace design and operation must necessarily remain an art. No amount of science can replace "know-how" ; it can only work in partnership.

The deviations of actual furnace systems from the simple ones to which theoretical treatment can be given are of two essential kinds:

(1) The fact that the *system is not isolated* into a single simple system. In the furnace chamber a number of different types of events are occurring simultaneously. These include, in general, the following : (a) the fuel is being continuously supplied and burns throughout a large volume of the furnace ; (b) heat is being transferred from hot to cooler gas, from hot gas to charge and to walls, and from walls to charge ; (c) gas flows through the system with complicated patterns of flow distribution. In fact, all the matters which were discussed in detail in Chapters 3, 4 and 5 have to be considered *simultaneously*. By neglecting the aerodynamic factors it is possible to set up simultaneous equations taking account of the effect of heat transfer and combustion upon each other. As has been mentioned in Chapter 4, this type of calculation has been carried out by Wohlenberg<sup>7.1</sup> for pulverised coal firing, by Heiligenstadt<sup>7.2</sup> for gas-fired continuous furnaces, by Mayers<sup>7.3</sup> for fuel beds and by Thring<sup>7.4</sup> for the open-hearth furnace. In each case, in spite of simplifying assumptions, the mathematical solution has required very complicated functions and calculations, and such calculations are only justified when the simplifying assumptions that have to be made can be chosen with sufficient knowledge of the system to make them exactly the right ones.

(2) *Impossibility of exact specification.* In all cases, moreover, some of the events in the furnace system are too complicated to be

7.1 WOHLBERG, W., and WISE, G.: "Distribution of Energy in the Pulverised Coal Furnace," *A.S.M.E. Trans.*, p. 531 (1938).

7.2 HEILIGENSTADT, W.: *Warmetechnische Rechnungen für Bau u. Betrieb von Ofen*, Chapter VIII (Düsseldorf, 1935).

7.3 MAYERS, M. A.: "Temperatures and Combustion Rates in Fuel Beds," *Trans. A.S.M.E.* (May 1937).

7.4 THRING, M. W.: "General Theory of Heat Transfer in the Open-hearth Furnace," *I.S.I. Special Report No. 37*, p. 171 (1946).



exactly expressed by any formula, for example, the effect of ash on refractories, the change in dimensions of the furnace system due to the appearance of deposits and dust, and the variations of radiating properties of a luminous flame in which active combustion is taking place.

In this chapter the manner in which the scientific results discussed in the previous chapters can be made use of in furnace design and operation in spite of these limitations will be discussed. This chapter thus consists of an attempt to show the interaction of the artistic side of the furnace problem with the scientific side. The allowance which must be made for the limitations of the scientific method depends upon one's objective in attempting to study what happens in furnaces, and it is therefore useful to classify the types of objective. This can be done according to the radicalness of one's aim in regard to the furnace, the first process involving minor changes to a single furnace and the last, development of a new industrial unit. The groups of a convenient classification are thus:

- (1) The study of a single furnace which has already been built with the aim of improving the operation of this individual furnace.
- (2) The study of a type or class of furnace to improve the type.
- (3) The design of a modification of existing types and the development of the fundamental laws governing the behaviour of such types.
- (4) Designing and developing a quite new type.

These four aims may be illustrated by the analogy given in Chapter 1 between the furnace and the living creature. The first objective is that of the general practitioner or veterinary surgeon, who has to find out what is wrong with an individual patient and make recommendations how to cure him. The second objective is that of the statistical biologist, who studies measurements of a species in order to gain knowledge of that species as a whole. The third is that of a medical research worker, who does experiments on specimens in order to find out how the species as a whole can be improved: for example, by removing their appendices. The fourth might be compared to the aim of the eugenist or breeder who wishes to develop a new variety. Each of the remaining sections of this chapter deals with one of these four objectives.



## 7.2. Diagnosis of a Single Furnace

**7.2.1. The Aim of Diagnosis.** In Section 1.1 the factors which constitute the success of a furnace system were enumerated, namely a good product, low fuel consumption and low capital and maintenance costs. It will frequently happen that a furnace which has been built will not be satisfactory in one or other of these three respects. The diagnosis of a furnace which has already been built is a method of measuring in a single case the extent of the disagreement between theory and practice, discussed in Section 7.1. Just as in medicine, it is in the case of the individual patient that the artistic side is strongest and the methods of scientific generalisations have least value. The simplest method of diagnosis of the trouble is the eye of the experienced man who looks in through spyholes at the flame and the charge. Such experience is, however, limited to the diagnosis of furnaces on which experience is available and is therefore limited, particularly in regard to new designs. There are three scientific methods of diagnosis which can supplement this one, namely, the use of probes, the quenching method and normal instrumentation of the furnace.

**7.2.2. Probes.** Probes correspond to the X-ray and the stethoscope of the doctor. They enable one to "see" what is going on inside, and particularly to split up the system into parts so that the approximate location of a fault can be ascertained. A number of probes have been discussed in different chapters, but it will be useful to give a survey of them at this point to show how far exact knowledge of what is going on inside the furnace can be attained.

(a) *Gas-sampling probes.* A water-cooled or refractory tube inserted into the furnace through the wall enables gas to be withdrawn from a point in the main stream to be analysed. In the case of refractory probes, if the gas contains both combustible and oxygen there will be reaction in the probe and the gas analysis will not correspond by the extent of this reaction to the actual conditions in the furnace. Where this is important, as for example in studying the course of the reaction in a diffusion flame or in a fuel bed, water-cooled probes (see Section 3.3.4) are to be recommended.\*

\* The method of cooling by the use of two separate tubes carrying the out and return flow of the water, whether concentric or not, is the one which has been found to be satisfactory because all partitions inside a single tube tend in practice to allow short circuiting unless they are welded all along on both sides.



The bore of the innermost tube (i.e. the sampling tube) should be large ( $\frac{1}{2}$ –1 inch) in cases where the main objective is to obtain continuous operation without maintenance, but should be small ( $\frac{1}{8}$ – $\frac{1}{16}$  inch or even smaller) in cases where the avoidance of secondary reactions is of special importance. It is convenient in cases where dust is liable to block the tube to have the offtake at right angles and to use a screwed plug or rubber tube<sup>7.4</sup> as a closure for the end of the inner tube, so that it can be removed and a cleaning rod inserted.

It is rarely satisfactory to insert these probes only a short distance through the wall of the furnace: they must either be inserted to a distance which is found by separate experiment to give the average, or the sample must be taken from a region where the gas is well mixed. In cases of stratification a number of readings at different distances must be taken, or a multiple probe, as discussed in Section 4.3.4, must be used. The design of gas sampling lines and especially of apparatus for taking uniform average samples over a long period of time is discussed in Chapter 32 of "The Efficient Use of Fuel". The use of stainless steel or silica linings to the water-cooled probe is desirable where the condensate is likely to be corrosive and electric heating of the inner tube to maintain it just above the dewpoint of the gases is desirable where soot or condensible hydrocarbons are likely to block the tube.

(b) *Gas-temperature probes.* These have already been discussed in Section 4.1.3. Their most important purposes are to enable the heat balance of the whole furnace system to be split up into parts when used in conjunction with the gas-sampling probes and to study flame length. This measurement of the gas temperature, together with a knowledge of the heat capacity of the gases and their quantity (both of which can usually be calculated) and their composition enables the percentage of the fuel energy carried out from one chamber to another by the hot gases to be directly worked out. Where the purpose is the splitting up of the heat balance the sensible heat meter described in Section 4.1.3 gives a more direct reading of what is wanted than does temperature measurement.

(c) *Wall- or charge-temperature probes.* For measuring the inside temperature of the furnace walls or of the charge, an optical pyrometer sighted on them through a small hole or partly open door may be used, subject to the two conditions: (1) there must be no visible radiation from the gas situated between the object

7.4 NICHOLS, L. H. F.: *J. Inst. F.* (Dec. 1940).



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sighted and the instrument. (2) The object sighted must either be effectively black or have a high emissivity at the working wavelength of the pyrometer\* or be surrounded by other objects at nearly the same temperature. The errors which will occur in the use of an optical pyrometer in cases where the surroundings are cold and the object sighted upon has a known emissivity are given in Fig. (7)1. A discussion by Reid and Cory<sup>7.5</sup> shows that a total-radiation pyrometer gives smaller errors than an optical pyrometer in cases where the sighted body is colder than its surroundings and *vice versa*.

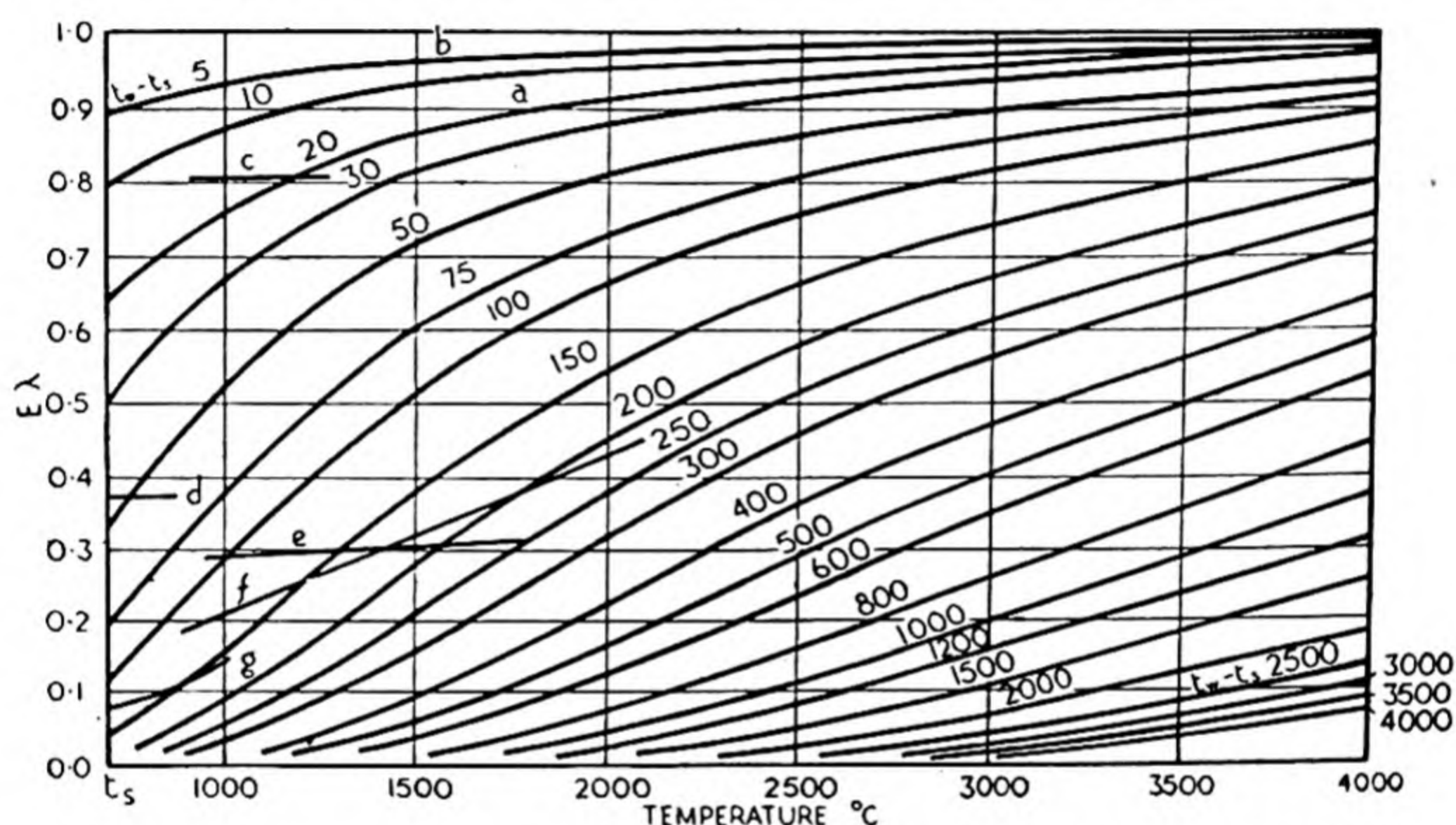


Fig. (7)1. Corrections to Optical Pyrometer Readings for Various Surface Emissivities (cold surroundings).

$E\lambda$  emissivity of object at wavelength of pyrometer.  
 $t_w$  true temperature of object.  $t_s$  observed temperature.

Total-radiation pyrometers may also be used for measuring the wall or charge temperature, but in this case errors arise if the gas between radiates even in the invisible part of the spectrum, so that it must not contain appreciable quantities of  $\text{CO}_2$  or  $\text{H}_2\text{O}$  at a temperature different from that of the body whose temperature is being measured. A well-known example of the use of such total-radiation pyrometers is in measuring the inside temperature of the crown in an open-hearth furnace. The instrument is sighted through a small hole in the back wall, arranged so that only the air at the top of the flame lies between it and the crown. Furnaces are also frequently equipped with total-radiation pyrometers sighted on closed end tubes in the wall or roof. Where such tubes are

\* Usually a red filter giving a wavelength of about  $0.65\mu$  is used.

<sup>7.5</sup> REID and CORY, *Combustion* (1944).



flush with the inside walls on their inner end, the inside wall temperature cannot be deduced from the reading of the instrument without separate calibration against a direct measurement of the inside temperature, for example, with an optical pyrometer sighted through the furnace, owing to the temperature gradient through the end of the tube. Where the sighting tube is carborundum and projects a distance of several diameters into the inside of the furnace, a reading approximating to the mean radiant furnace temperature can be obtained if convection from the flame is not severe. In general such a tube gives a useful measure of the conditions in the furnace, but its temperature does not exactly correspond to that of any other part of the system.

Thermocouples let into holes in the brickwork or in the charge provide a convenient method of measuring the temperature at the point where the tip is situated, base metal couples being suitable for temperatures up to  $1100^{\circ}\text{C}$ . provided the sheaths are gas tight, and noble metal (platinum-rhodium) up to  $1550^{\circ}\text{C}$ . In order to measure surface temperatures inside the furnace it is usually necessary to insert a number of couples at different distances from the surface and extrapolate their readings to zero distance.

(d) *Heat flow probes.* These have been discussed fully in Section 4.5.2, the wall type of instrument being, when situated opposite to a cold target, convenient for use for measuring the true flame radiation, while the inserted type can be used as a research tool for measuring the net heat transfer from flame and walls to the charge as a function of position in the furnace. The wall type sighted through the flame at the opposite wall is convenient as a measure of the radiating power of the flame, since the wall temperature is primarily governed by the flame radiating power. Such an instrument can usefully be calibrated in terms of black-body radiating temperature, but its temperature reading will not of course correspond to the true flame temperature.

(e) *Static pressure measurements.* These can be simply made in a furnace by inserting a refractory tube through a hole in a brick as discussed in Section 5.1.3.

(f) *Gas velocity and quantity.* The use of pitot tubes for measuring the velocity of flow of the hot gas at any point in a furnace system has been discussed in Section 5.4.4, while orifices for measuring the total quantity of gas and the dilution method for the same purpose were discussed in 5.4.2 and 5.4.5 respectively.

(g) *Probes for measuring the mixing length or turbulence.* The



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use of a radio-active gas for indicating the degree of turbulence in the furnace system was discussed in Section 5.7.3.

(h) *Solid or liquid sampling.* Many methods have been developed for withdrawing samples of solids or liquids from chosen points in the furnace system. Thus in open-hearth practice it is customary to insert "spoons" into the molten slag and steel and to pour the resulting spoonful of liquid into a mould to give a sample for seeing how the process is going at the given point. Similar arrangements have been made to withdraw samples of dust, powdery materials and deposits from a plant while it is running. One instrument, designed for measuring the rate of deposition of dust in a gas-producer main, consists of a disc on the end of a wire, which was lowered on to the surface of the deposit and, after exposure for a given time, covered with a small cone slid down the wire. The disc, with a sample of dust protected in this way, was removed and weighed.

(i) *Instruments for measuring the shape and free area of channels.* In actual practice many parts of a furnace system become more or less blocked up with powdery or partly fused deposits. Calculations of the flow conditions which do not take account of these blockages are frequently greatly in error, and the only way to correct this error is to insert measuring rods into the system and feel for the boundary of the deposit.

Other types of probes can in special cases be of value in furnace systems, as for example, where the electrical conductivity of the flame is to be measured. Extensive use of electrical probes has, however, been confined to flames such as that of a Bunsen burner,<sup>7.6</sup> and they have not been used in actual furnaces.

**7.2.3. The Diagnosis of Furnace Faults from Probe Measurements.** The general methods of using the results of probe measurements can be discussed in relation to three ways in which a furnace can deviate from the ideal.

(a) When a furnace produces a product which is irregular either from point to point *across* the discharge stream or with time *along* it, it may be concluded that the heating cycle is not being controlled in a satisfactory way. The best probes to use in this case are those which measure the charge temperature and the heat transfer to the charge. Where the output is irregular in time, instruments measuring the temperature and analysis of the gas leaving the

<sup>7.6</sup> KISSELMANN and BECKER: "Electrical Conductivity of Flame," *Ann. d. Phys.*, p. 49 (1936).



heating chamber can also throw light on the cause. In these cases the probes simply show the location of the irregularity in the heating chamber or at what time it occurs. Having obtained this information the irregularities must be removed by applying the laws of gas flow, combustion and heat transfer discussed in Chapters 3, 4 and 5 to find some method of removing it.

(b) When a furnace is considered to be using too much fuel, probes can be used to construct a heat balance of greater or lesser refinement, i.e. to find where each fraction of the heat is going. The first step in constructing a heat balance is to make a gas balance, i.e. to assess the quantities of gas flowing into the system and out of it at different points. This is done by the methods discussed in Section 5.4. It is also possible to replace measurement of the total quantity of gas flowing by measurement of the quantity of fuel supplied, together with measurement of  $\text{CO}_2$  content of the gases at various points. The quantities of air and combustion gas can be calculated from these measurements by the methods discussed in Section 3.1. Knowing the quantities of gas flowing at various points in the system, the sensible and chemical energies of these gases may be calculated as discussed in Section 2.1 and the heat balance constructed as in Section 2.1.2. Heat losses through the furnace skin can be assessed by difference, but they should be compared with the values calculated, as discussed in Section 4.4. Having constructed a heat balance the main source of loss of heat can be at once indicated, and the best way to set about reducing the fuel consumption can be deduced. In this connection it is, however, important to bear in mind the fact, pointed out in Section 2.2, that heat losses from a high-temperature region are much more significant in increasing fuel consumption than those from a low-temperature region. Excessive fuel consumption will usually be traced either to (1) excessive air infiltration ; (2) inadequate insulation ; (3) inadequate heat-absorbing area for the required output ; or (4) too long a flame.

(c) The third cause for dissatisfaction with a furnace is excessive capital and maintenance costs. These can arise in two ways : if the furnace has excessive wear on the refractories or if the furnace has an unduly low output. Excessive wear on refractories is generally due either to excessively localised combustion or, more usually, its converse, an excessively long flame, or to irregularities of gas flow, such as a jet or flow restriction causing impingement of flame on the refractories. To see whether too long or too short a



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flame is the cause, gas analysis probes can be used as discussed in Section 4.3.4, while irregularities of gas flow can be discovered by measuring the distribution of velocity by the methods discussed in Section 5.7. Another way of discovering localised combustion is by the use of the heat-flow meter. This instrument can be used to give a direct quantitative measure of the properties of a flame in a furnace by taking readings with it at the two ends of the furnace and deducing the percentage drop in flame radiation along the flame. Thus a long flame in an open-hearth furnace can give a higher reading at the outgoing than at the ingoing end, while a short one gives a drop in radiation of 10–20%. The heat-flow meter can also be used to give warning of the possibility of excessive wear due to a reversing furnace operating unsymmetrically on the two half-cycles to measure the distribution of the flame heat across the width of the furnace and to indicate the exact direction and location of abnormal heat intensities.

When a furnace is found to have an abnormally low output the cause is usually one of three things, viz. :

- (1) the furnace gives a badly controlled product when the output is raised ;
- (2) it is not supplied with, or cannot burn, sufficient fuel to give a greater output. In this case it simply will not heat more than a certain output to the required temperature. Remedies involve increasing the area of the combustion ports and the flues ;
- (3) the heat transfer rate from the flame to the charge is insufficient, so that even when sufficient fuel is burnt it merely carries its extra heat out of the furnace. This will be shown by a drop in the thermal efficiency or output fuel ratio as the output is increased ; it can only be overcome by increasing the surface area of the charge or the heat transfer coefficient, by shortening the flame or by adding preheaters.

**7.2.4. The Quenching Method.** In addition to the insertion of probes into a hot furnace, a method which is frequently used for diagnosis, particularly on smaller furnaces, is to shut down the whole system and cool it in such a way that the high-temperature conditions are not disturbed significantly during the cooling. This method has been applied to the study of the location and nature of the ash released in fuel beds and gas-producers, where the bed must either be quenched in steam or inert gas or else sealed off so that no



combustion can take place during the cooling. In the case of large industrial gas-producers the bed<sup>7.7</sup> when cooled has been sectioned by a man with a spade, and the ash and types of coke at different places directly viewed.

The quenching method is also of particular value for studying the conditions of the refractories at various points in a furnace system. Samples taken from the crown of open-hearth furnaces have enabled the conditions in the bricks to be very carefully studied.<sup>7.8</sup> For the study of refractories it is not usually necessary to take special precautions about the atmosphere during the cooling down, but allowance must be made for changes of crystal structure during this period or it may be necessary to cool the sample rapidly to prevent such changes.

Finally, the conditions of the charge in shaft or bath type furnaces can usefully be studied by shutting down and examining it from place to place when it is cold. In the case of bath furnaces this method can only be applied to the smaller units, since the removal of the frozen material is too costly on the larger scale.

**7.2.5. Normal Furnace Instrumentation.** In addition to the probes which may be inserted into the furnace to diagnose the cause of trouble it is being found increasingly important to equip all industrial furnaces of any size with an adequate system of instrumentation to enable the day-to-day variations in operation to be checked and the furnaces to operate at the optimum conditions for fuel economy, accurate processing and long life, even though the required output rate and quality may be changed.

In the heavy chemical industry, up to 6% of the capital cost of plants may be spent on instrumentation, and it is now clear that industrial furnaces would benefit by at least an equal proportion of the total expenditure. The blast furnace, however, is probably an exception in this connection, as 6% would represent a figure far in excess of the sum required. While instruments will never completely replace the trained eye of the experienced operator, they can certainly greatly relieve the load of responsibility upon him. Instruments leave even the most experienced operator free to give much more attention to the material being processed in the furnace, as he requires to give relatively less attention to keeping his furnace in order and looking after fuel economy.

The problems of instrumentation are very varied in different

<sup>7.7</sup> NOYES, E.: Private communication.

<sup>7.8</sup> HARVEY, F. A.: *J. Amer. Cer. Soc.*, **18**, No. 3, p. 86 (Mar. 1935).



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furnace processes, according to whether, as in the open-hearth furnace, the system operates nearly at the limiting temperature for the materials, or whether, as in the glass tank furnace and blast furnace, the main problem is to ensure thorough and complete formation of the liquid product with minimum fuel consumption. It is therefore of some interest to compare proposals for complete instrumentation schemes on different types of furnace systems. If instrumentation is extended to include automatic control, the interest of the comparison becomes even greater.

Broadly speaking, all the automatic control systems applied to furnace systems correspond to four main automatic linkages more or less interlocked.

The first of these is control of the fuel input rate so as to obtain a given temperature at one or more places in the furnace to protect the furnace refractories and to ensure accurate processing of the charge. Ideally, of course, this control would be arranged to provide a given temperature in the batch material itself, but where this is impossible the control is usually based on a wall temperature measurement.

Secondly, automatic control of the fuel/air ratio, either to a given value or, preferably, to provide a given  $\text{CO}_2$  content in the waste gases. This is the control which is primarily concerned with fuel economy, the aim being to reduce the vast amount of heat which is used to heat unnecessary air in furnaces, an effect which becomes more serious the higher the temperature of the furnace. Automatic control of the fuel/air ratio by flame radiation<sup>7.9</sup> measured in the manner discussed in Section 4.5.2. has not yet been attempted, but this might provide a practical method of enabling the ratio to be adjusted to give the highest possible flame temperature, since reliable automatic control by gas analysis has not yet been achieved. In the blast furnace control of the fuel input is carried out by charging weighed amounts of coke, and control of the blast can really be regarded as the primary control.

In a number of cases, particularly the open-hearth furnace, automatic control of the furnace pressure is carried out by adjustment of the stack damper or waste-heat boiler fan linked to a pressure-measuring element. The aim of this control is really to assist the second one by reducing the amount of false external air that leaks

7.9 THRING, M. W.: "Relation of Heat Transfer to Successful Furnace Design and Operation," *Iron and Coal Trades Rev.*, p. 1063 (Nov. 1948).



in and so upsets the fuel/air ratio, which can only be controlled by the air entering in the proper way. Finally, various special controls can be incorporated, of which the most important is automatic control of reversal of regenerators operated either on a time cycle or on the temperature of the flue gases leaving the appropriate regenerator.

(a) *Scheme for blast-furnace instrumentation.* A complete scheme for blast-furnace instrumentation is illustrated in Fig. 7(2).<sup>7.10</sup> In this scheme, the hot-blast stoves, operating as they do with heat from fuel gas burned within them and not, as in the open-hearth furnace, with waste gases, have to be controlled and instrumented as completely as a separate furnace installation requiring a record of the volume of fuel gas, control of the air/fuel ratio, and control, or at least warning, of the dome temperature. The temperature of the combustion gases leaving the stove which is being fired and of the hot blast leaving the stove which is heating air are also recorded, the latter assisting judgment in changing over from one stove to another. A complete scheme of automatic control of changeover is not yet advocated, but is an obvious future development.

Metering the quantity of air supplied to a blast furnace presents special problems, since it is often difficult to separate the air supply to the different blast furnaces of a group, and subsidiary meters for the supplies to each one are clearly desirable in such cases. The temperature of the preheated air going to the furnace itself is controlled by bleeding in unpreheated air direct from the blast main. A compensated meter is being developed by the Ironmaking Division to carry out this process automatically.

Turning to the supply of the solids into the blast furnace, instrumentation here has to be very elaborate, involving a complex interlocking and warning scheme to ensure that the right quantity of materials is charged in the right order at the right time. Methods for making the shape of the stockline in the top of the furnace visible are being developed since it has been shown that segregation of the charge materials is largely effected by the contour of the stockline.

So far as the blast-furnace gas leaving the top of the furnace is concerned, there is room for further development in the use of CO and CO<sub>2</sub> recorders and in improving the sampling lines to

<sup>7.10</sup> RIDGION, J. M., and GILLINGS, D. W. (Ironmaking Division of British Iron and Steel Research Association.) Private communication.



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operate them reliably, spot analysis being mainly used at present. The gas temperature and humidity are recorded at each stage of the

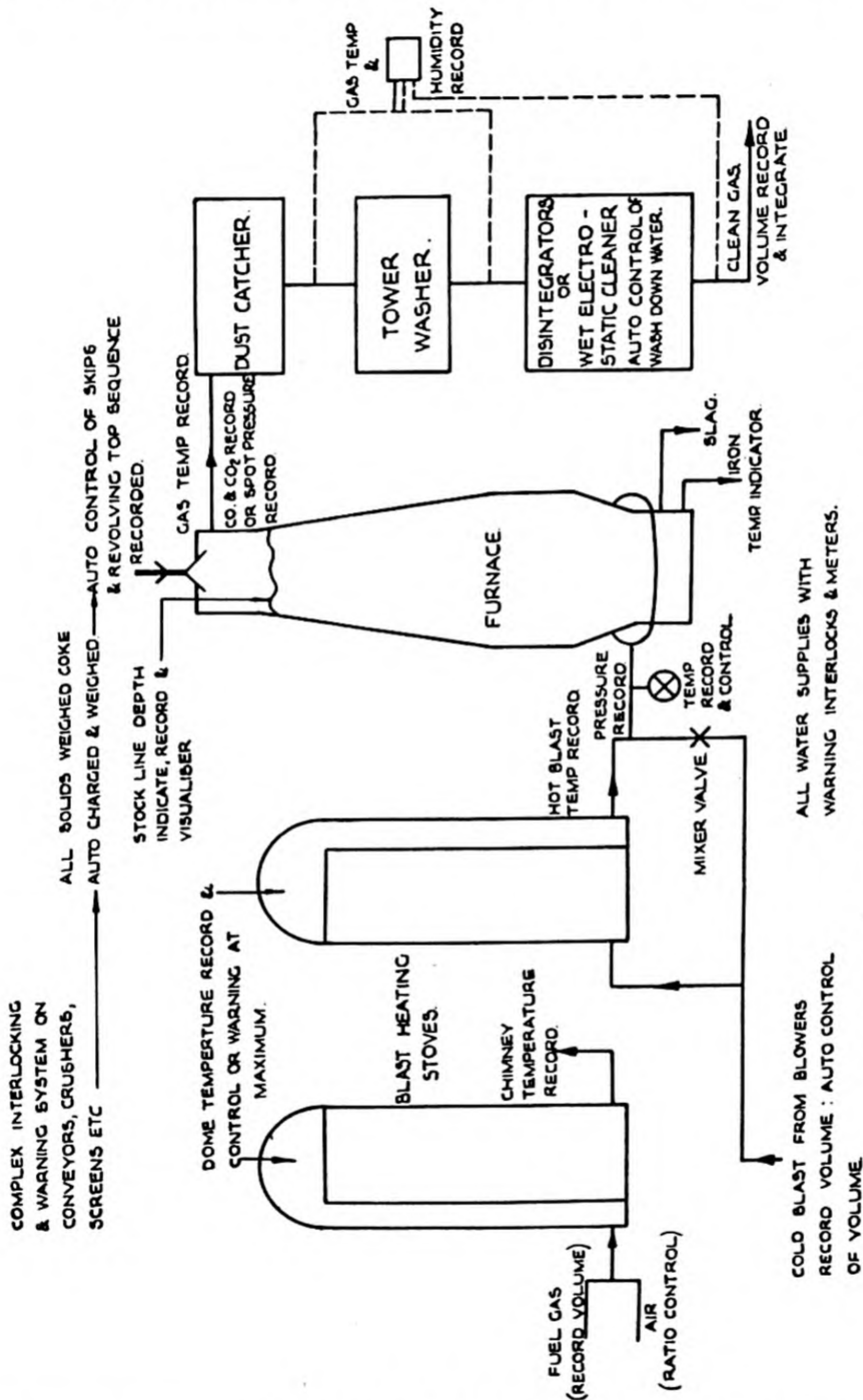


Fig. (7)2. Schematic Diagram of Blast Furnace Instrumentation.

cleaning process, and, when this is completed, its volume can be recorded and integrated accurately. This measurement is very



necessary, since the blast-furnace gas output is a large and important factor in the economics of the furnace, nearly half the calorific value of the fuel burned appearing as chemical energy in the gas.

(b) *Open-hearth furnace and glass-tank furnace.* The open-hearth and glass-tank systems may be taken together, since they are both fired with regenerative producer gas or oil flames from alternate sides, the chief differences between them being that: (i) the steel furnace operates on a batch process with steadily rising temperatures, while the glass tank works on a continuous process; and (ii) the steel furnace operates much nearer the temperature limit so far as the refractory materials are concerned. Schemes of instrumentation for the two which have been proposed are shown in Figs. (7)3 and (7)4.

Various schemes for more or less complete automatic control of the open-hearth furnace and glass-tank furnace are under investigation, covering all the three controls already discussed in principle. Automatic control of producer gas supplied to the open-hearth furnace by the roof temperature has been successfully operated by R. C. Baker,<sup>7.11</sup> and a number of experiments with automatic control of fuel oil supply are being carried out in various steelworks. Oil is usually regarded as easier to control than hot raw producer gas. At present, air/fuel ratio control is being confined to controlling the ratio of the measured air volume flowing in through the air valve and the measured fuel supply, but this scheme is subject to the disadvantage that the large volume of air which leaks into the furnace is not adequately allowed for in this way, and improved methods based on analysis of waste gases, measurement of the volume of waste gases, or rendering the furnace system relatively airtight, are under consideration. An additional control which can be, but very rarely is, installed is the adjustment of steam/oil ratio. Automatic control of the pressure in the furnace is probably the oldest application of such control to open-hearth furnaces. Control of this variable is of great value because it has not yet been found possible to seal the charging doors, which frequently hang with a 4-inch gap all round. Hence, unless the furnace pressure is close to atmospheric, the amount of air which leaks in, or fuel gas which leaks out, is liable to be enormous. This is usually done by linking a pressure-measuring device, connected to a tapping on the furnace roof, to

<sup>7.11</sup> BAKER, R. C. : "Installation and Use of Instruments on Open-hearth Furnaces," *J.I.S.I.* 157, Pt. I, p. 81 (Sept. 1947).







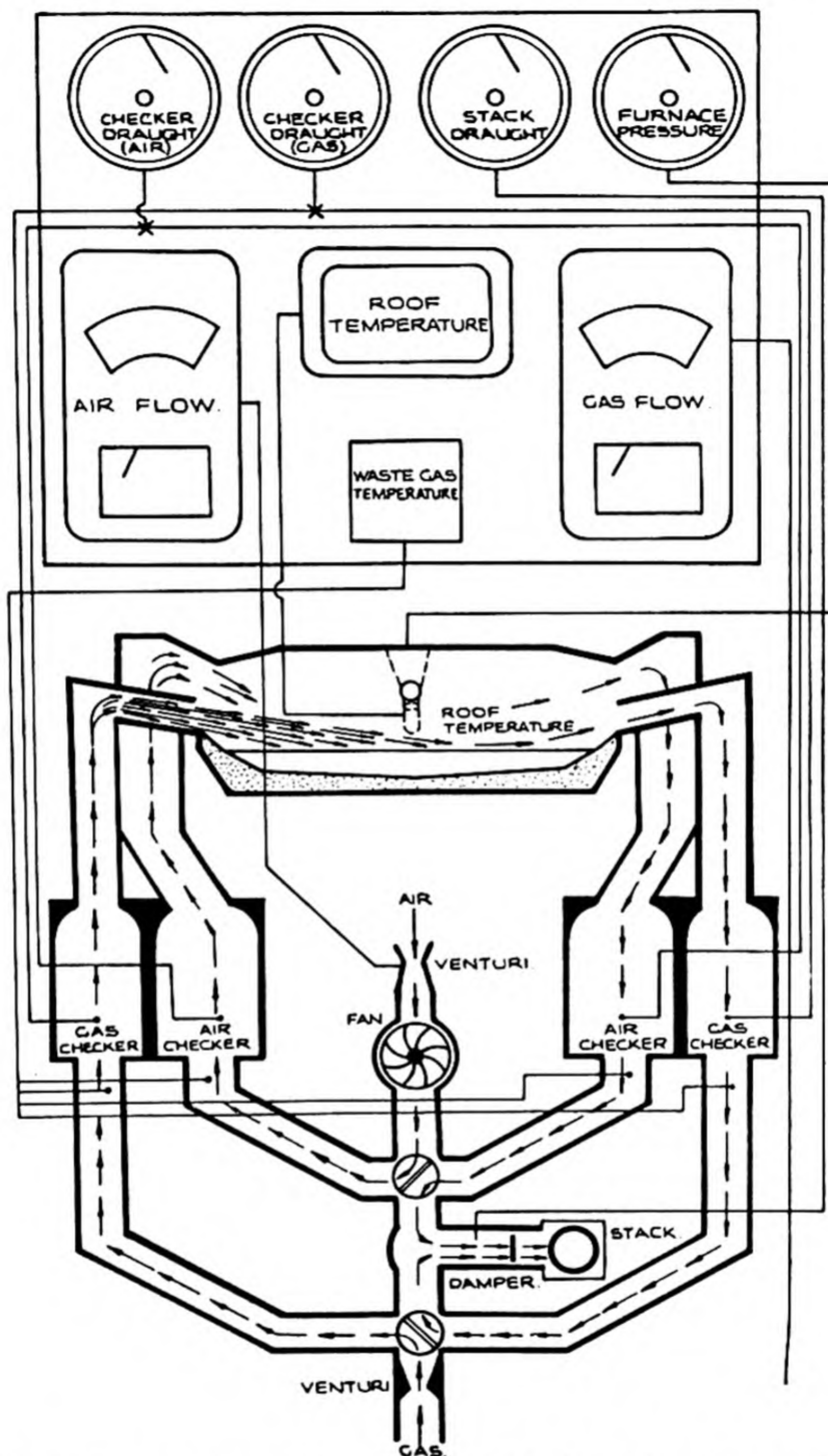


Fig. (7)4. Instrument Arrangement for Cold Pig Fixed Open-hearth Furnace. Producer Gas Fired.

(Courtesy of the United Steel Companies Ltd.)

the stack damper of the furnace. Such linkage has to contain special modifications to prevent hunting when the furnace doors are opened.



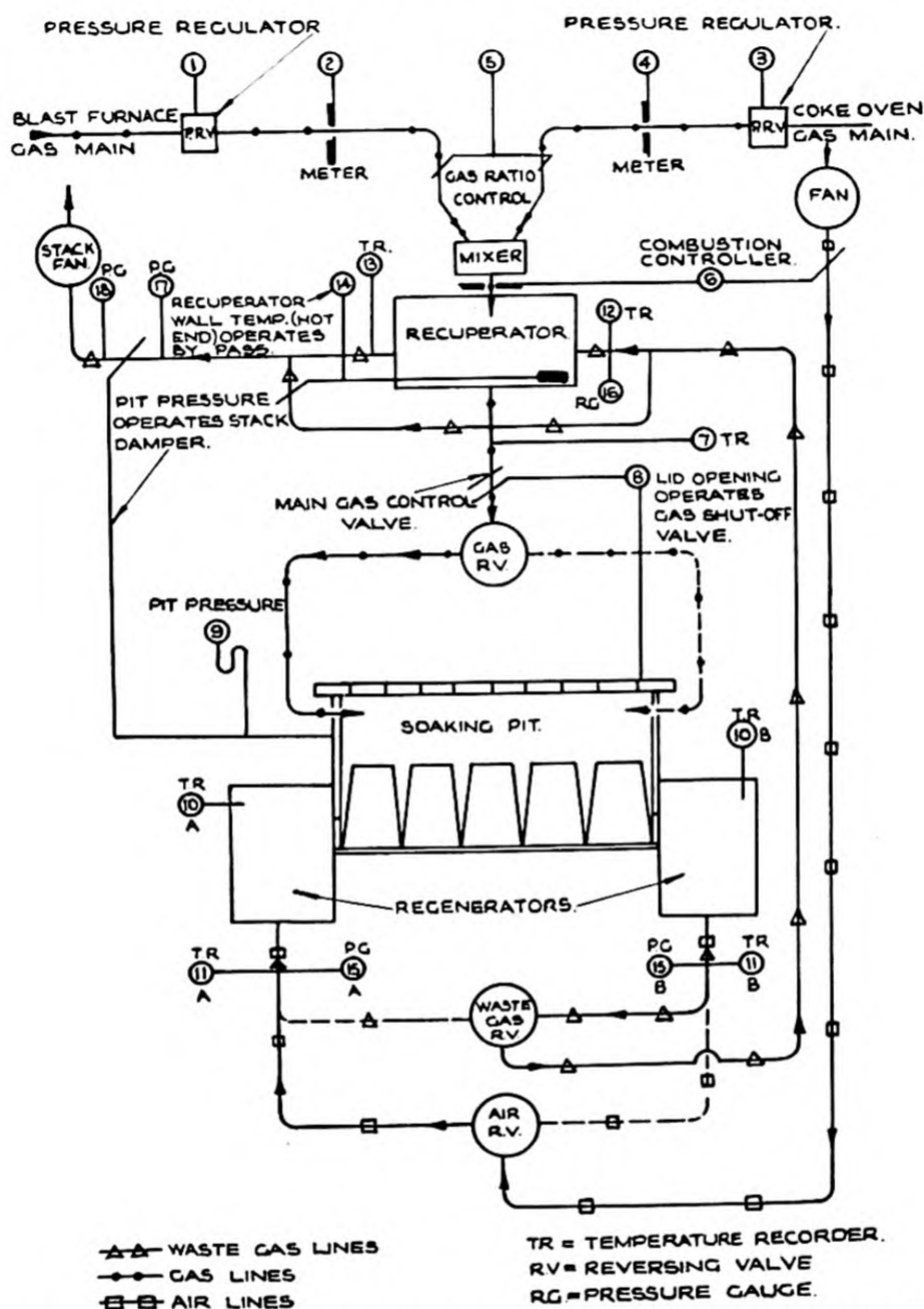


Fig. (7)5. Scheme of Instrumentation for Mixed Gas-Fired Soaking Pit with Air Regeneration and Gas Recuperation.

(c) *Soaking pit.* A proposed scheme of instrumentation for a mixed-gas-fired soaking pit with air regeneration and gas recuperation<sup>7.12</sup> is shown in Fig. (7)5, all the measuring points being numbered. In addition to the usual controls and measurements, the ratio between the blast-furnace gas and coke-oven gas used for firing is automatically adjusted to a predetermined value, the air/gas

<sup>7.12</sup> SAVAGE, L. H. W. (Plant Engineering Division of British Iron and Steel Research Association.) Private communication.



ratio being controlled by the combustion controller. It is hoped it will later be possible to add to this scheme combustion control by gas analysis and possibly also a device for controlling the fuel input, based on measurement of the steel temperature.

Another feature of special interest is the device whereby opening the soaking-pit lid to insert or withdraw an ingot operates a valve shutting off the fuel gas, this automatic valve being in series with the main gas-control valve.

(d) *Control of a Continuous Slab-heating Furnace.* A control and measurement scheme for a continuous slab furnace, 18 ft. wide and 80 ft. long, heating slabs by means of flames underneath and over them in the heating-up zone, and a flame over the top in the soaking

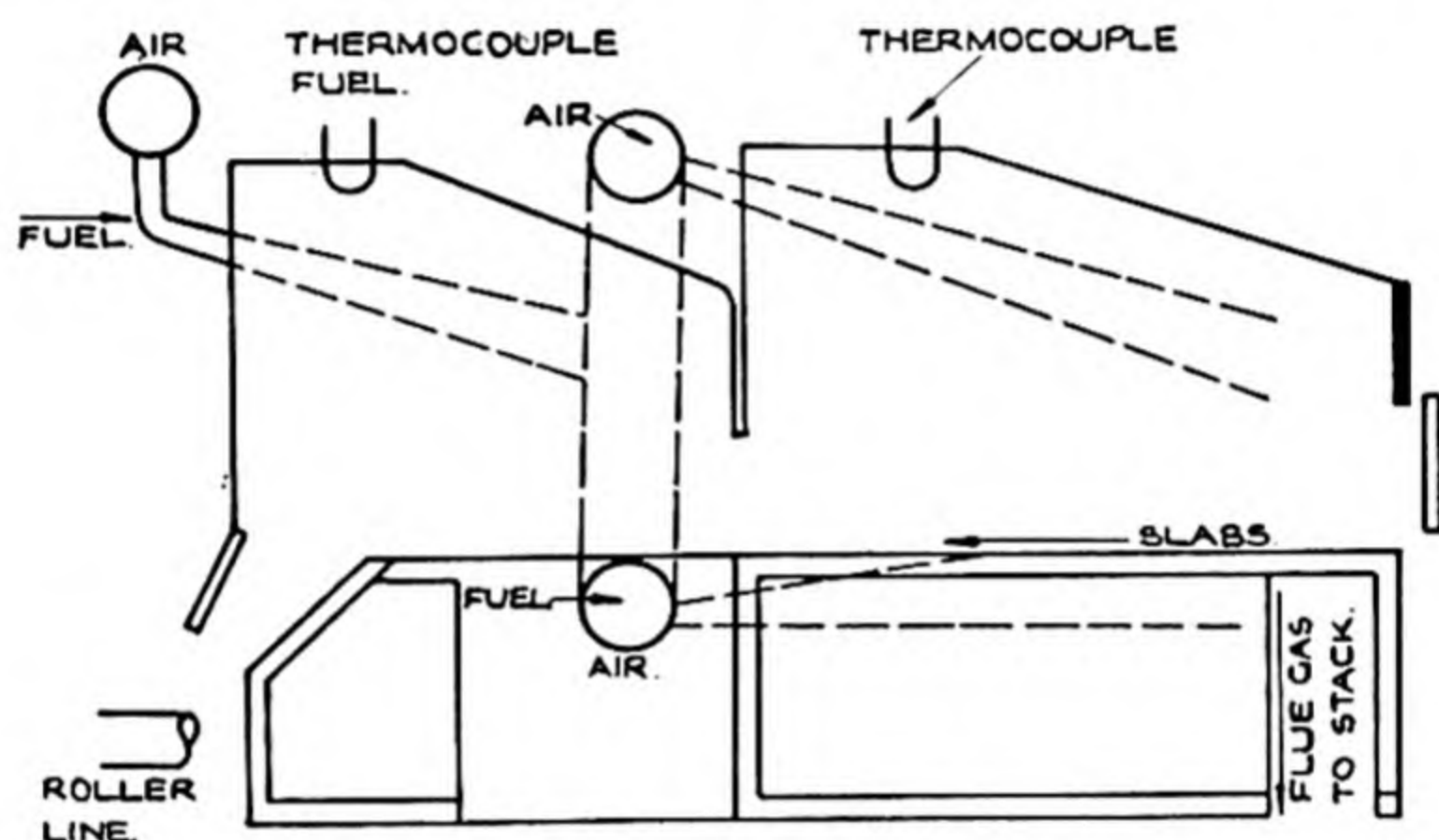


Fig. (7)6. Diagram of Slab-heating Furnace with a Top Zone, a Bottom Zone, and a Soaking Zone.

zone, has been discussed by C. E. Duffy,<sup>7.13</sup> and is illustrated diagrammatically in Fig. (7)6. The interesting feature of the furnace is that the fuel supply to the two burners heating the first zone is automatically controlled by a thermocouple, while the fuel supply to the second zone is controlled by a second thermocouple. The two burners to the first zone receive proportionate quantities of oil, maximum oil input to the bottom zone being 300 galls./hr., and to the top zone 360 galls./hr. The thermocouples used for automatic control are precious metal enclosed in sillimanite and carborundum tubes and the control is automatic by pneumatic controllers. The fuel/air ratio and the furnace pressure are also controlled automatically.

The examples which have been very briefly discussed above can

<sup>7.13</sup> DUFFY, C. E.: "Remote Control of Continuous Slab Heating Furnaces," *Iron and Steel Engineer*, 70 (Dec. 1947).



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be regarded as representing good modern practice for instrumentation and automatic control in various systems. A number of additional instruments can be installed for research purposes, but would not be required for normal operation. Considering the wide variety of materials being heated in these different furnaces, it is perhaps surprising that the instrumentation and automatic control has so many common features through all of them, until one considers that they are all concerned with the combustion of carbonaceous fuels with atmospheric air in order to produce heat.

### 7.3. Statistical Study of a Furnace Type

7.3.1. **The Scope of the Statistical Method.** This section deals with the problems of finding a satisfactory method of comparing the performance of furnaces which differ both as regards design and operation factors (whose effect is to be found to show which are the best) and output factors (factors depending mainly on the order-book of the firm for which correction must be made or which must be averaged out). The discussion is limited to furnaces which have a general type similarity, and it will be shown from the examples that in some cases furnaces producing quite widely differing products can be compared provided the type is the same.\* The ultimate aim of the comparison can be therefore either a formula which will give *a measure of the goodness of the furnace performance*† after correction has been made for the output factors,

\* It is necessary to clarify what is meant by saying that the type is the same. Thus, for example, (1) a comparison of a glass-tank furnace with an open-hearth furnace is possible<sup>7.14</sup> or (2), any flame-fired pot or crucible furnace with any other such furnace, or (3) of any gas-fired shaft furnace with any other, whereas a comparison of a member of one of these classes with a member of another would not enable one to say which fulfilled its function better, i.e. which was a more perfect design of its own type. An examination of these examples shows that type can be defined for the present purpose by saying that the thermodynamic heating arrangements (type of combustion, type of heat transfer, type of waste-heat recovery) and the physical conditions of the charge (smooth, solid, broken solid, solid in muffle, liquid in bath, liquid in crucible, liquid in tube, gas in tube) must be the same.

† The word "performance" could apply to any of the quantities discussed in Section 1.1, by which a furnace is assessed as good or bad, but fuel economy or some related quantity (thermal efficiency or no-load heat requirement) is usually taken. It could be equally possible, however, to take such a quantity as life of the crown or refractory cost per unit output, or output per unit area of bath for melting furnaces.

<sup>7.14</sup> JEBSEN-MARWEDEL, H.: "Present-day Glass Practice in Comparison with Steel Making," *Stahl und Eisen*, p. 858 (1942).



or a formula showing the effect of output factors and design and operation factors upon this performance criterion. These will be referred to as the first and second aims in the rest of this section. Examples of the two kinds of factors, sorted out for the first of these two aims, may be taken from a scheme which has been used for the comparison of glass-tank furnaces.<sup>7.15</sup> The output factors given there are :

- (1) The temperature of the several parts of the furnace.
- (2) The rate at which glass is withdrawn.
- (3) The furnace dimensions.
- (4) The standard of glass quality.
- (5) The colour and composition of the glass.
- (6) The furnace age.

While the design factors are :

- (1) Furnace design, including applied cooling or insulation.
- (2) Furnace operation conditions, such as pressures and combustion conditions.
- (3) Gas-producer design.
- (4) Gas-producer operation conditions.

This comparison of performance results on existing furnaces is essentially a statistical problem, since in practice several factors vary at once and there are always some (such as the psychological condition of the operators) which are important but cannot be measured. Now in some cases where insufficient data are available on a single furnace the deficiency can be partially remedied by collecting similarly insufficient data on such a large number of furnaces that the unmeasured factors can be averaged out. The method is therefore essentially different from that discussed in Section 7.2, where experimental means are taken to obtain a great mass of data about the processes going on at a given moment inside a single furnace.

There are three necessary steps in this application of statistics to furnace data, namely :

- (1) *Classification of the variables.* One or more quantities, which will be denoted as  $X$ , are chosen as the measure of performance, a number of others which will be denoted as  $Y, Z, W \dots$  are chosen as the significant output factors. The design and

<sup>7.15</sup> "The Assessment of the Thermal Performance of Tank Furnaces for Melting Glass," *J. Soc. Glass Technology*, **28**, p. 33 (1948).



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operation factors  $y, z, w$  are also tabulated ; they are not used in the formula when the final aim is to see which set of these gives the best value of  $X$ . The  $X$  factors are usually chosen from among the following four: quality of product, quantity of product from a given size of furnace, fuel efficiency and refractory costs.

(2) *Derivation of the formula connecting  $X$  with  $Y, Z, W$ .* This formula is usually assumed to be of the type  $X = \alpha Y + \beta Z + \gamma W + \alpha' y + \beta' z + \gamma' w$  and the operating results are used to calculate the coefficients  $\alpha, \beta, \gamma$ . This process is carried out by a combination of empirical statistical methods and of theory. The more fully the data can be used to give the formula by the former methods the more confidence can be placed in the result, since theory always depends on various assumptions.

(3) *Comparison of different furnaces.* When the formula is obtained the results ( $X$  values) of different furnaces can be corrected by means of it to standard values  $X_0$ , corresponding to standard values  $Y_0, Z_0, T_0$  of the output factors  $Y, Z, T$ , and hence the furnaces can be directly compared in spite of the differences in the output variables for the achievement of the first aim. This step is omitted for the achievement of the second aim.

In the next three subsections these steps will be discussed in turn, and illustrated by three examples which have been thoroughly worked, viz., the case of glass-tank furnaces<sup>7.16</sup>, open-hearth furnaces<sup>7.17</sup> and blast furnaces.<sup>7.18, 7.19</sup>

**7.3.2. Specification of the Variables.** The choice of the relevant output variables in a furnace system is primarily a matter of common sense ; indeed, statistics can do very little more than test and make quantitative the conclusions of common sense as to the effect of these variables. It may happen, however, that in the course of the statistical investigation the importance of variables which were not expected to have any effect will appear, in which case it will be

7.16 Report on the Furnace Committee of the Soc. Glass. Tech.: *Trans. S.G.T.* (1943).

7.17 ROBERTSON, F. L., and THRING, M. W.: "A Statistical Analysis of the Output of an Open-hearth Furnace," *J.I.S.I.*, **162**, p. 31 (Sept. 1949).

7.18 LECKIE, A. H.: "The Study of the Thermal Performance of Open-hearth Furnaces by the Correlation of Operating Data," *J.I.S.I.*, No. 1, **149**, pp. 419P-442P (1944).

7.19 EVANS, E. C., and BAILEY, F. J.: "Blast-furnace Data and Their Correlation," *J.I.S.I.*, No. 1, p. 53 (1928); SWAN, A. W.: "The Work and Organisation of a Statistical Department in Heavy Industry," *J.I.S.I.*, **160**, p. 1 (Sept. 1948): blast-furnace analysis on pp. 16-17.



necessary to collect further data in which the extra variables are also tabulated. Statistics can also be very useful in disproving or proving "old wives' tales"—that is to say, verifying whether some of the rather startling facts which are often believed to hold really do so.

As illustration of the variables, the three examples quoted make the following choice.

(1) *Open-hearth furnace.* Performance variable used by Leckie,  $X$ =thermal efficiency, i.e. the ratio of the heat/hr. supplied to the charge, to the heat/hr. supplied to the furnaces compared for a number of furnaces. Performance variable used by Robertson and Thring,  $X$ =output in tons per hour on a single furnace. Output variables used by Leckie are :

- (1) period during the week ;
- (2) air/gas ratio ;
- (3) total rate of heat input ;
- (4) period in the furnace life ; and
- (5) size of furnace.

Robertson and Thring found it necessary to introduce intermediate variables, such as the rate of acceptance of oil fuel by the furnace, the rate of oxidation and the carbon content of the charge at melt down.

(2) *Blast furnace.* Performance variable  $X$ , used by Evans and Bailey, =lb. of carbon per ft.<sup>2</sup> of hearth and per hour ;  $X$  used by Swan, output per ton of coke (i.e. thermal efficiency) corrected by assumed factors for coke, iron and ore quality. Output variables used by Evans and Bailey :

- (1) hearth diameter ;
- (2) slag weight ; and
- (3) silicon content of iron.

(3) *Glass-tank furnace.* Performance variable, lb. of standard coal per ft.<sup>2</sup> of glass surface and per hour, corrected to zero output ; output variables :

- (1) furnace temperature (governed by the type of glass being made) ;
- (2) rate of glass production/ft.<sup>2</sup> ;
- (3) furnace area ;
- (4) colour of glass (affecting heat absorption properties) ; and
- (5) age of furnace.



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It will be seen that in nearly all cases the performance variable  $X$  is connected with the fuel efficiency, i.e. the statistical method is really being used in each case to give a measure of the fuel economy of the furnace. The other two features which assess the success of the furnace, discussed in Section 1.1, are not dealt with. The actual way of assessing the thermal efficiency is, however, different in each case. Leckie<sup>7.18</sup> uses, directly, a figure intended to be the thermal efficiency. Evans and Bailey use lb. of carbon and lb. of iron/ft.<sup>2</sup> of hearth and per hour, so that taken together lb. of carbon/lb. of iron can be obtained. This is not, however, corrected for the different heat requirements of the different irons, so that their results are not directly the thermal efficiency. The Society of Glass Technology have found that in glass tanks the weight of fuel per unit weight of glass is so strongly affected by the rate of withdrawing of glass from the furnace that they have considered it better to give the weight of fuel per unit weight of furnace area corrected to zero output. In other words, they have decided that the fuel economy of furnaces cannot be compared at all by comparing the fuel per unit of output in their industry, but only by comparing the amount of fuel necessary to keep the furnace up to temperature when there is no output.

**7.3.3. Construction of the Formula.** The formula connecting  $X$  with  $Y, Z, W$  and with  $y, z, w \dots$  is generally constructed by a mixture of theoretical considerations and statistical correlation. This is necessary because there are generally insufficient data to test statistically for all the factors, a process which would require many readings taken over the whole range of each variable. The theoretical idea which is accordingly most usually imported to supplement the empirical method is the use of the first law of thermodynamics applied to the furnace chamber itself. This gives :

$$\begin{aligned} \text{Heat required} = & \text{heat lost through the walls of the furnace} \\ & \text{chamber} \\ & + \text{heat supplied to the charge} \\ & + \text{heat carried out of the furnace chamber by} \\ & \text{the hot gases} \quad \dots \dots \dots (7/1) \end{aligned}$$

All these heat requirements must be expressed on the same basis, e.g. per unit area of bath or per unit weight of output and per unit time. To use this formula it is necessary to take account of the fact that as the output is raised so the heat carried away by the hot gases is also raised, and hence the increase in the second term on the



right-hand side does not represent the whole effect of increasing the output.<sup>7.20</sup> It is usual when the theoretical calculation is being made to assume that the heat lost through the furnace walls is independent of the output and hence is equal to the heat loss when there is zero output. The heat carried out of the furnace by the hot gases is allowed for not by a separate term but by additions to the first two terms; thus the heat actually lost through the furnace walls is replaced by the total heat put into the furnace chamber when there is no load (i.e. the heat put into the walls plus that in the exit gases at no load) and the heat actually required by the charge  $(W \cdot H_o)^*$  is replaced by this heat  $W \cdot H_o$  multiplied by a factor  $\gamma$  which allows for the efficiency of heat utilisation in the chamber. This gives the formula:

$$\text{Heat required} = \text{No-load heat input} + (\text{heat supplied to charge}) \cdot \gamma \quad (7/2)$$

For example the Society of Glass Technology corrects for the furnace output by introducing a term for the heat to the charge divided by a heat availability factor, which in the absence of other data is taken as 0.55, i.e.  $\gamma = 1/0.55$ .

In the other two cases the factor  $\gamma$  is determined empirically from the results by the method of statistical correlation, but the general form of the formula is still based on the first law of thermodynamics. This is as far as theory can usefully be applied. The other coefficient in the formula must be evaluated by statistics. The general method of statistical correlation is discussed very clearly in Lecture 5, p. 43, of a course of lectures entitled "Statistical Methods in Industry" given by L. H. Tippett and published by the Iron and Steel Industrial Research Council.<sup>7.21</sup> In a controlled experiment the effect of a single variable is found by maintaining all the others constant and altering this one over the desired range. In the type of investigation with which we are concerned this is not possible, and in general all the variables are changed from one reading to the next. The first step is to plot the performance variable  $X$  against one of the output variables, say  $Y$ . In general, this will produce a "cloud" of points which will show a

\* Where  $W$  = output in lb. per unit area and per hour ;  
 $H_o$  = heat required by 1 lb. of charge.

7.20 THRING, M. W. and REBER, J. : "The Effect of Output on the Thermal Efficiency of Heating Appliances," *J. Inst. Fuel* (1945).

7.21 TIPPETT : *Statistical Methods in Industry* (Iron and Steel Ind. Research Council).



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more or less systematic trend (see 7.21 Fig. (7)7). Such a cloud is known as a “scatter diagram.” A visual examination of the scatter diagram gives at once a qualitative measure of the effect of this variable  $Y$  on  $X$ . It also enables readings which are exceptional to be spotted and, if there is an independent reason for doing so, eliminated. To obtain quantitative results from the scatter diagram

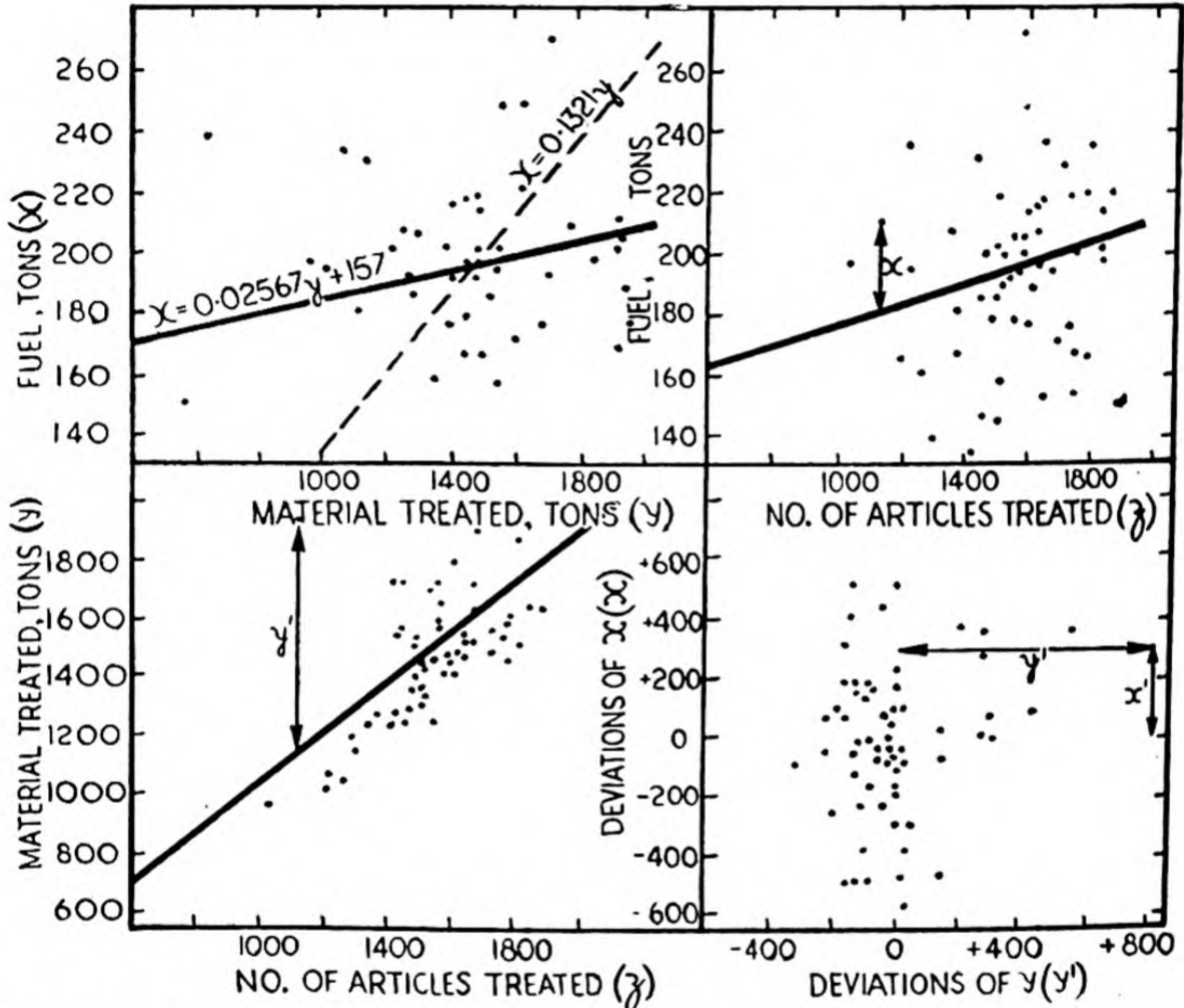


Fig. (7)7. “Scatter Diagram” of the Performance Variable  $X$  Plotted against Output Variable  $Y$ .

$$V_{xy} = +0.2432; V_{yz} = +0.6810; V_{xz} = +0.2657; V_{xyz} = +0.088.$$

the variation of  $X$  with  $Y$  can be calculated as follows. First the mean of all the values of  $X$  (denoted  $\bar{X}$ ), and the mean of all the values of  $Y$  ( $\bar{Y}$ ) are calculated. Then, if the relation between  $X$  and  $Y$  can be taken as linear, it is given by the formula :

$$X - \bar{X} = a_{xy} (Y - \bar{Y})$$

where

$$a_{xy} = \frac{\Sigma(Y - \bar{Y}) \Sigma(X - \bar{X})}{\Sigma(Y - \bar{Y})^2} \quad \dots \quad (7/3)$$



An alternative but less accurate method of calculating the relation between  $X$  and  $Y$  which reduces the amount of calculation, and also does not depend on the assumption that the relation between  $X$  and  $Y$  is linear, is the group method. Leckie makes use of a convenient way of carrying out the group method in which all the relevant data for each case, i.e.  $X$  and the output variables  $Y, Z, W$ , are written out on a single card. When a hundred or more of such cards are available they are arranged in order of increasing values of the particular output variable, say  $Y$ , which is being studied. They are then divided into groups of approximately equal numbers of cards such that there are at least six groups with not less than fifteen cards in each group. The average value of  $X$  and  $Y$  from the cards in each group is then calculated and plotted one against the other, a single point being thus obtained for each group. Where a punched card tabulating machine is available this type of calculation can be done automatically.

How far the apparent valuation of  $X$  with  $Y$  found by the above methods is genuine and not due to the random effect of other variables must be tested by calculating the "correlation coefficient." This is given by the formula :

$$\frac{\Sigma(Y - \bar{Y})(X - \bar{X})}{\sqrt{[\Sigma(X - \bar{X})^2 \Sigma(Y - \bar{Y})^2]}} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (7/4)$$

If  $X$  were a linear function of  $Y$  *alone* all the points would lie on a single line and the correlation coefficient would have a value of  $+1$  or  $-1$  according to whether the slope of the line was positive or negative. If  $X$  and  $Y$  are completely independent the points will be randomly scattered but the calculated correlation coefficient can have by chance a value slightly different from zero. To determine whether the "old wives' tale" that  $Y$  affects  $X$  is likely to be true, it is usual to take as a test whether the correlation coefficient calculated by (7/4) from the observed values is greater than the value which it would have only one in twenty times by accident if there was in fact no correlation. The values at this one in twenty level can be calculated from the laws of chance and depend on the number of observations. With five observations the correlation only stands a chance less than one in twenty of being accidental if the coefficient exceeds 0.88, with ten observations 0.63, with twenty 0.44, and with sixty 0.25.<sup>7.22</sup> If the correlation coefficient lies below the value

<sup>7.22</sup> FISHER, R. A., and YATES, F.: *Statistical Tables for Biological Agricultural & Medical Research*, Table VI. (Oliver & Boyd, 1948.)



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appropriate to the number of observations it is not possible to make a definite statement for or against the tale.

Using these methods, Evans and Bailey calculated a statistical formula connecting the carbon consumption in lb./hr./unit area of hearth of a blast furnace with the output of molten iron expressed in the same units. From the linear relation of these quantities they concluded that there is an optimum output at which the fuel efficiency is greatest. Leckie found that the thermal efficiency when plotted against the rate of heat input to the open-hearth furnace is constant for lower values and falls off for higher ones. The Society of Glass Technology in their final formula concluded that the efficiency rises with output without limit over the range considered. It has been shown<sup>7.20</sup> that in fact all furnaces will at first give a rising efficiency as the output is raised from zero because the bulk of the fuel is consumed to keep the furnace to temperature. At very high outputs, however, the gas will begin to leave the furnace at a temperature considerably in excess of the furnace, and hence the thermal efficiency will once again begin to fall. The blast furnaces considered by Evans and Bailey were evidently working in the region of the critical thermal efficiency, the open-hearth furnaces considered by Leckie above this region, and the glass-tank furnaces below it. Boilers tend in general to be operated above the region and high-temperature furnaces below it.

The effect of the various variables  $Y$ ,  $Z$ ,  $W$ , etc., is usually calculated by the methods of statistical correlation, on the assumption that they can be added together linearly to give the formula of the type  $X = \alpha Y + \beta Z + \gamma T$ . Such an assumption is made by Evans and Bailey, who obtained the final formula :

$$\text{Carbon consumed/ft.}^2 \text{ hearth-hr.} = \frac{500}{D} + XI + 0.28S \quad (7/5)$$

where  $D$  is the diameter of the hearth in feet (this factor allows for the heat loss necessary to maintain the furnace at the full temperature) ;

$X$  is the heat requirement of 1 lb. of iron measured in lb. of equivalent carbon ;

$I$  is the output measured in lb. of iron/ft.<sup>2</sup> hearth-hr. ; and

$S$  is lb. of slag/ft.<sup>2</sup> hr.

The coefficients of the first two terms on the right-hand side were found by the statistical method.  $X$  was found to have a mean



value of 0.56, but to be related to  $T$  (the time in the shaft in hours) by the formula :

$$X = 0.67 - 0.022T \quad . \quad . \quad . \quad . \quad . \quad (7/6)$$

The coefficient of the last term in 7/5 was calculated on theoretical grounds and was assumed to be constant in all cases.

The final formula obtained by the Society of Glass Technology<sup>7.23</sup> is

$$Q = 0.026 \left( \frac{GH}{AK_A K_T} - \frac{W_c}{AK_A} \right) \quad . \quad . \quad . \quad . \quad . \quad (7/7)$$

in which  $Q$  = quantity of heat required per second to maintain 1 ft.<sup>2</sup> of glass surface at 1400° C., B.Th.U./ft.<sup>2</sup>/sec. ;

$G$  = total quantity of fuel consumed in the furnace per 24 hrs., tons/24 hrs. ;

$H$  = the net heat content of the fuel, B.Th.U./lb. ;

$A$  = total area of the glass in the melting and working portions of the furnace exposed to the crown when the furnace is new, ft.<sup>2</sup> ;

$W$  = weight of glass discharged from the furnace per 24 hrs., tons/24 hrs. ;

$c$  = quantity of heat required, at the reversing valve of the furnace system, or at an equivalent position, to melt the raw materials needed to yield 1 lb. of glass and to raise its temperature to 1400° C., B.Th.U./lb. ;

$K_T$  = numerical correction to be applied when the furnace, as a whole, is working at some temperature other than 1400° C., derived theoretically ; and

$K_A$  = numerical factor to enable correction of furnace area to be made to the standard of 600 ft.<sup>2</sup>, derived theoretically.

Recently, as part of the general study of three open-hearth furnaces, called  $H$ ,  $K$  and  $S$ , to compare the port designs, a statistical analysis was carried out on about 1 year's data.<sup>7.24</sup> From each of the three furnaces in this paper, the weekly averages and the variabilities of the charging plus melting and refining times over

<sup>7.23</sup> Furnace Committee : "Assessment of Thermal Performance of Tank Furnaces," *J. Soc. Glass Technology*, **28**, p. 37 (Feb. 1944).

<sup>7.24</sup> BOSWELL and THRING : "The Variability of Furnace Performance," *I.S.I. Special Report No. 37*, Section I, Pt. 2.



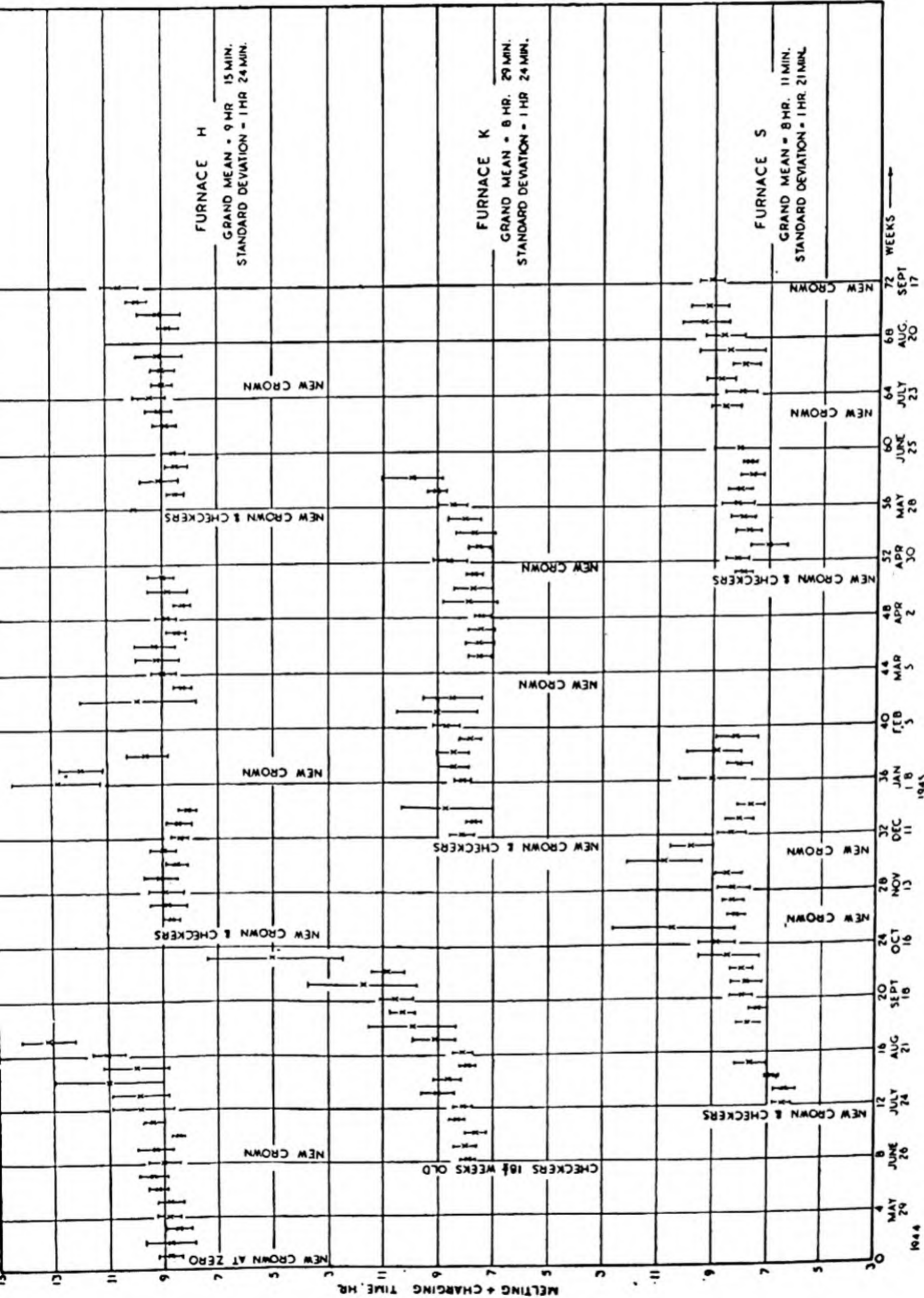


Fig. (7)8. Weekly averages and standard deviations of melting and charging time for three Open-hearth Furnaces  
Plotted against Furnace Life.  
(Courtesy of the Iron & Steel Institute.)



about a year and a half on each of the three furnaces are calculated and presented as functions of the data, age of crown and checkers, steelmaking variables, fuel supply and other factors. It is shown that the variability is so large (80 minutes) that reliable comparisons cannot be made between furnaces in normal operation unless data for at least sixty casts are available, but that when sufficient data are available there are consistent differences of an hour in the mean melting plus charging times between *S* furnace (fastest) and *H* furnace. *K* furnace is only 20 minutes slower than *S* on the melting plus charging time, but is as slow as *H* on the refining time, for which both are 20 minutes slower than *S*. Fig. (7)8 shows the variations of weekly averages of charging plus melting times with history for the three furnaces. Evidence is

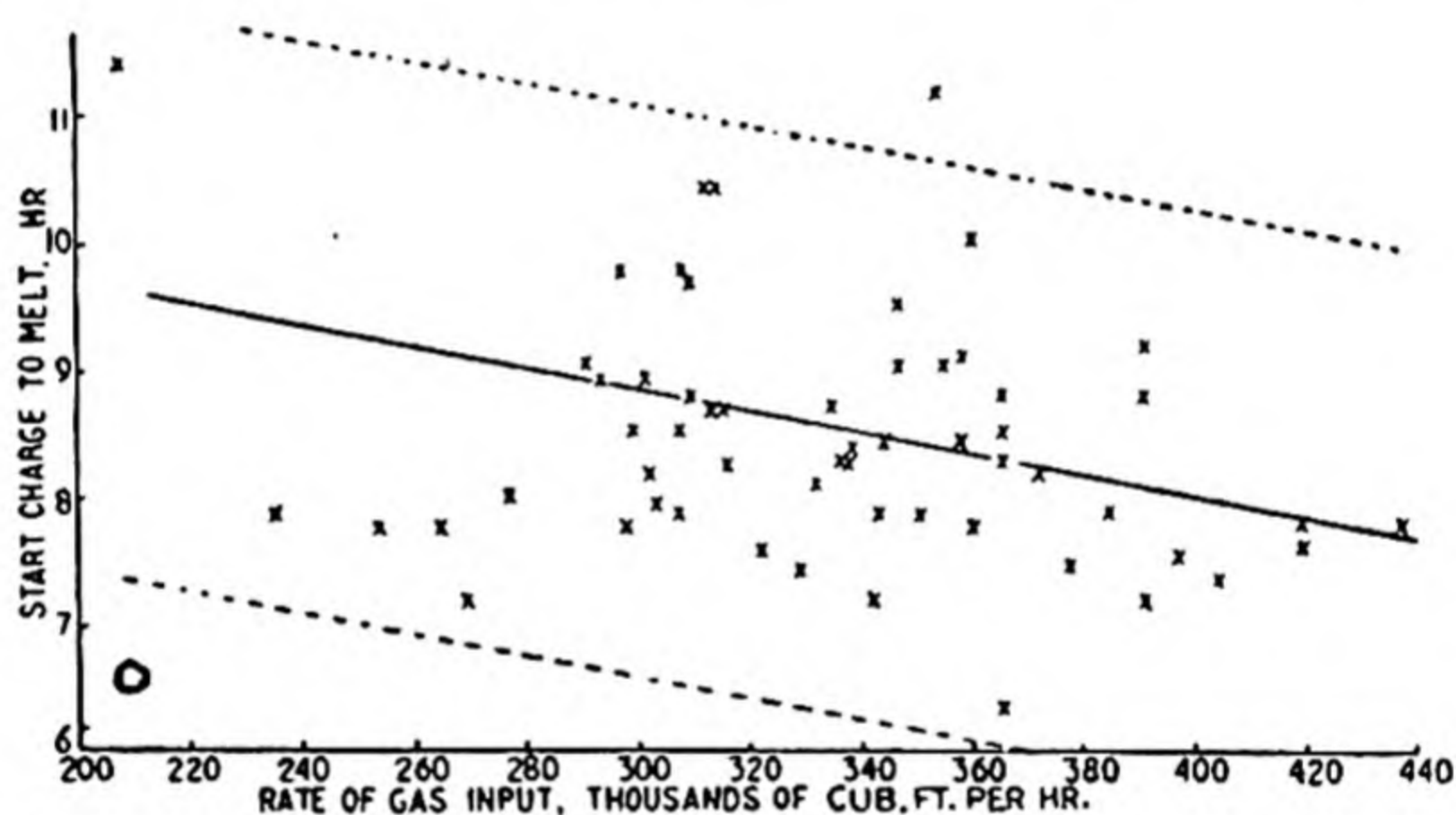


Fig. (7)9. Regression of Time to Melt on Rate of Gas Input, Furnace *S*.

given that, like the melting time, the charging time is probably largely dependent on the flame characteristics. Except when crown and checkers are old, the variability from cast to cast within the same week nearly equals the total variability, but both the variability and the mean values are found to increase significantly when the furnaces are old. None of the steelmaking variables showed a significant correlation with the melting plus charging time, whereas both the port area and the gas-supply rate accounted for a significant part of the cast to cast variability. Fig. (7)9 shows the relation between the time to melt and the average rate of gas input to *S* furnace over a number of casts. The correlation shown here was found to be just significant. It is concluded that the flame radiating power is the major factor by which improvements in output can be expected, and an indication is found that metering



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the gas may alone be sufficient to produce an appreciable improvement. The later paper by Robertson and Thring,<sup>7.17</sup> giving a statistical analysis of casts on a single furnace making low sulphur low carbon steel, indicates that the furnace output is mainly affected by the rate at which oil fuel can be burnt in the furnace without melting the roof and the rate at which the bath is oxidised during melt-down.

**7.3.4. Use of Formulæ to Compare Different Furnaces.** Having obtained a formula by the methods discussed in 7.3.3, the final step in using statistics to compare furnace designs is to collect data on many different furnaces, correct them to standard values of the output variables and see which ones then have good values of  $X_o$  and which bad. The three examples discussed here are of interest in that the blast furnace and open-hearth furnace operate in all cases with materials whose thermal properties (apart from the actual known heat requirements) are almost constant, whereas the Glass Society has been more ambitious and has compared furnaces whose actual heated material is varied both as regards working temperature and heat absorbing properties. The Glass Society has accordingly had to draw considerably more on theory for calculating the coefficients in the formula. Nevertheless, it does appear that their final formula can be used to say that, for example, a certain furnace melting glass at  $1600^{\circ}\text{C}$ . is of inherently better design than a larger one melting glass at  $1300^{\circ}\text{C}$ . because the value of " $Q$ ," their symbol for  $X$ , after correction for the temperature and area differences, is lower ( $X_o$  in the notation of 7.3.1).  $Q$  is thus the fuel heat requirement/ $\text{ft.}^2$  surface-second, corrected to a standard glass of  $1400^{\circ}\text{C}$ . and a standard bath area. They have laid down the figures that values of  $Q$  in B.Th.U./ $\text{ft.}^2$ -sec. under 5 correspond to a "good" furnace, between 5 and 6 to a "fair" furnace, between 6 and 7 a "poor" furnace and over 7 a "bad" furnace. They have in particular used the formula to find the variation in effective performance of a furnace with age, and in one case the figure for a furnace has been shown to rise steadily from  $4\frac{1}{2}$  on the new furnace to  $5\frac{1}{2}$  after 60 weeks' running.

Leckie has not expressed his results in the form of a final formula, but since steels vary very little in melting temperature he has been able to plot the efficiency directly against various variables; for example when the output is taken as  $Y$  he finds that certain furnaces are taking less than the optimum quantity of gas and others more;

<sup>7.17</sup> ROBERTSON, F. L., and THRING, M. W.



when the air-gas ratio is taken as  $Y$ , one such furnace shows the highest efficiency at an air-gas ratio of 2.2 to 1 (since the theoretical air-gas ratio is 1.3 to 1 this shows the necessity for considerable excess air in open-hearth furnaces of conventional port design). He has also shown that there is a considerable drop in efficiency at the beginning of the week, due to the week-end cooling, recently eliminated in many open-hearth shops, and a marked variation in efficiency throughout a campaign. The work is not, however, at a stage where it is possible to conclude that one furnace is of a "good" design and another of a "bad" design.

From their formula, Evans and Bailey draw the conclusion that the furnaces with the greatest effective time of contact between the reducing gases and the ore have the least fuel consumption, but that the time of contact should be increased by other methods than reducing the rate of driving, since this also increases the relative heat loss. It thus appears that the application of the statistical method to a furnace type can be of considerable value to the designer and operator to tell him the importance of different factors. There is no doubt that a considerably greater use of this method will be made in the near future.

#### 7.4. Designing a Modification of an Existing Type

**7.4.1. The Use of Formulæ.** In this section we are concerned with the problems of a designer who has built a certain type of furnace in the past, but wishes to design a new furnace of this type, either larger or intended to be an improvement on existing designs. He may wish to incorporate improvements in the form of new materials which will allow higher temperatures to be used, or to improve the shape so as to obtain a better heat distribution or thermal efficiency. In fact, of course, nearly all furnaces which are built are modifications in this sense. In cases where the calculation of the effects of the altered factor is a straightforward application of formulæ given in Chapters 2 to 6, as for example increasing the convection velocity of the flame or the size of the furnace, the furnace can be designed quite accurately from calculations, and in all cases such calculations are recommended, with one very important proviso. This is, *that the simplifying assumptions are well chosen*. It is in the choice of the simplifying assumptions that the "art" of furnace design comes in. It has been shown in 7.1 that it is not possible to calculate out in detail all the simultaneous equations by which the different branches of knowledge



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specify what will happen in a furnace. Consequently it is always necessary to make definite and clear-cut simplifying assumptions, but these must not assume the very thing one sets out to improve, for example the thermal efficiency.

The most usual simplifying assumptions for such calculations are that one knows the flame temperature in the furnace, the temperature of the inside of the brick walls and the flame emissivity. From these assumptions the rate of heat transfer to the charge and the wall heat losses can be calculated from the formula of Chapter 4 for any given size of furnace; hence the temperature of the gases leaving the furnace and the heat carried by them out of the furnace can be calculated and thus a rough thermal balance set up for the furnace chamber. The percentage of heat returned by recuperators or regenerators is usually taken as standard, but here again if one of the objects of the new calculation is to design a furnace with improved recuperators or regenerators, these must be calculated out in detail as discussed in Sections 4.3.4 and 4.5.4.

Having constructed an approximate picture of the thermal conditions in the furnace in this way, the aerodynamic conditions are then calculated from the quantities of gas flowing and the temperatures, and care is taken that the channels to convey the gases at various points are large enough to give reasonable pressure drops after making due allowance for possible deposits in operation.

Finally, the fuel-burning arrangements are designed to enable the calculated amounts of fuel to be burnt with air at the calculated preheat temperatures according to the methods given in Chapter 3, using the mixing formula of Chapter 5 if the firing is to be by diffusion combustion.

**7.4.2. Model Tests.** In cases where the similarity conditions can be expressed whereas the actual details are too complicated to be solved mathematically, model tests may be of great value both in order to perfect existing designs and to test new ideas. There are three well-known fields in which models have proved of value, namely:

(1) *Aerodynamics* where the Reynolds number is the similarity criterion. This is discussed in detail in Section 5.7.4.

(2) *Forced and natural convection to special shapes.* It has been shown in Section 4.2.3 that the convection to various simple shapes can be calculated from well-established formulæ based on experiments using these shapes. Where the heat transfer to a special shape, such as a "finned" surface or a packing, is of especial



importance, it is not very accurate to apply existing formulæ, but it is possible to carry out model tests to give a measure of the heat transfer which can be expected.

(3) *In unsteady state heat flow* electrical and hydro-dynamic models have been developed for solving the complex differential equations, since the flow of electricity and water can be made to obey identical differential equations by a suitable choice of the arrangement.<sup>7.25</sup>

Models can also be used in more complex systems to study the processes in a full-size furnace where two or more processes proceeding simultaneously are important by studying similar processes in a miniature furnace,<sup>7.26</sup> but in these cases a very careful investigation of the similarity conditions is necessary to ensure that the results are applicable to the full scale.

**7.4.3. General Laws Governing a Combustion-heated Furnace.** One part of the work of the medical research worker is to find the fundamental laws governing the behaviour of the types of individual he is studying, since such laws set a limit to the improvement which can be expected. An attempt has recently been made in a paper by Thring and Reber<sup>7.20</sup> to establish such laws in general form for any system heated by combustion gases. It is, of course, well known that in such a system it is necessary to put in fuel at a certain minimum rate to bring the system up to a working temperature before any output at all is obtained. As the fuel input is gradually increased, the thermal efficiency at first rises from zero. In this paper the very simplest analysis is made of such a heating system regarded as a hollow tube, the walls of which are being heated by heat transfer from the gases so that the heat in these cases splits into three parts which:

- (1) heats the charge ;
- (2) maintains heat losses through the walls ; and
- (3) is carried out of the heating system by the flue gases.

<sup>7.25</sup> JACKSON, R., SARJANT, R. J., WAGSTAFF, J. B., EYRES, N. R., HARTREE, D. R., and INGHAM, J.: "Variable Heat Flow in Steel," *J.I.S.I.*, **11**, p. 211P (1944).

PASCHKIS, V., and BAKER, H. D.: "A Method for Determining Unsteady-state Heat Transfer by Means of an Electrical Analogy," *Trans. A.S.M.E.*, **64**, p. 105 (1942).

MOORE, A. D.: "The Hydrocal," *Ind. Eng. Chem.*, **128**, p. 704 (1936).

<sup>7.26</sup> THRING, M. W.: "The Construction of Models in which more than one Process is similar to the Original," *Trans. Inst. Chem. Eng.*, **26**, p. 91 (1948).



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From these assumptions it is shown that there is a certain maximum efficiency obtainable above which the rise in heat carried out by the flue gases is greater than the fall in the relative heat loss through the walls.

A simplified heating system is shown in Fig. (7)10, while typical curves for such systems are shown in Fig. (7)11. The general conclusions may be expressed in the following formulæ :

$$\eta = (1 - b \cdot \overline{1 - a})(1 - e^{-1/z}) - \frac{ab}{z} \quad \dots \quad (7/8)$$

$$\frac{H_w}{H_a} = \frac{a \cdot b}{z} \quad \dots \quad (7/9)$$

$$\frac{H_b}{H_a} = (1 - b \cdot \overline{1 - a})e^{-1/z} + b(1 - a) \quad \dots \quad (7/10)$$

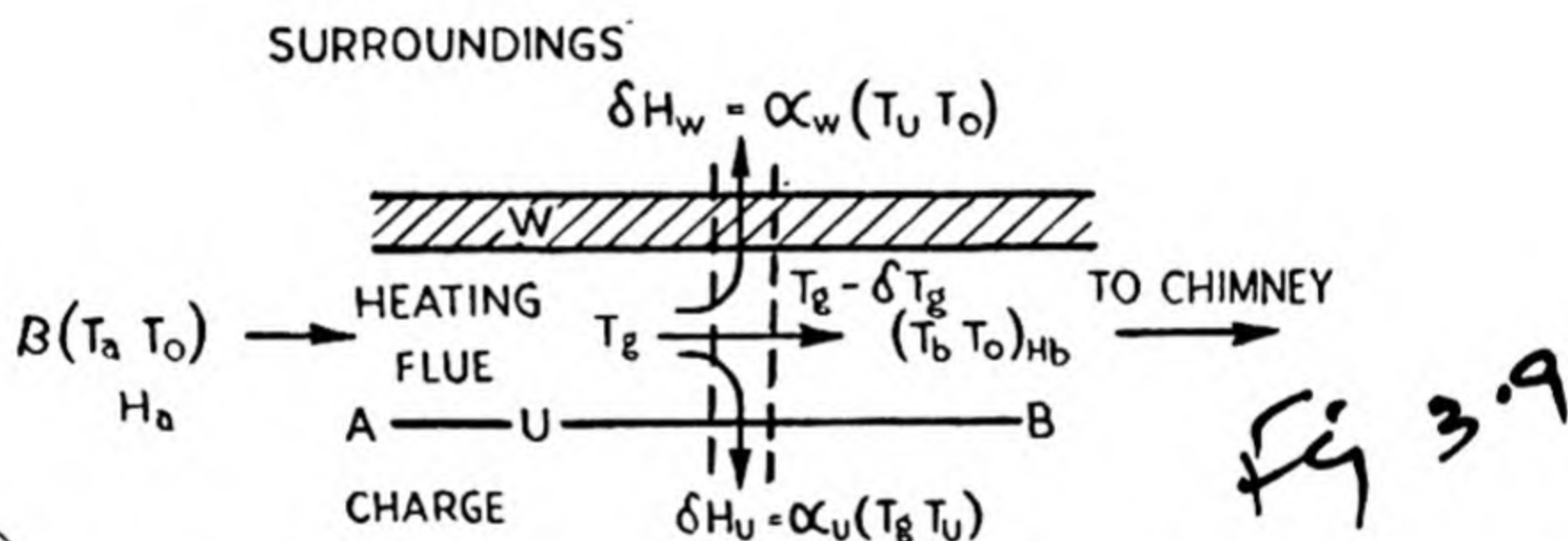


Fig. (7)10 Diagrammatic Sketch of the Heating System used for Calculation.

In these formulæ

- (i)  $H_w$  is the heat lost through the walls per unit time (C.H.U./hr.);

$H_a$  is the total heat input carried into the channel by the flame gases in unit time (C.H.U./hr.);

$H_b$  is the heat carried out of the furnace in the waste gases (C.H.U./hr.)

$H_u$  is the useful heat to the charge in unit time (C.H.U./hr.).

so that 
$$H_a = H_w + H_b + H_u$$

- (ii)  $\eta$  is the thermal efficiency, i.e. the ratio between the heat  $H_u$  transferred usefully to the charge and the heat  $H_a$  in the fuel, so 
$$\eta = \frac{H_u}{H_a}$$
 From formula 7/8 it follows that  $\eta$  rises to a maximum and



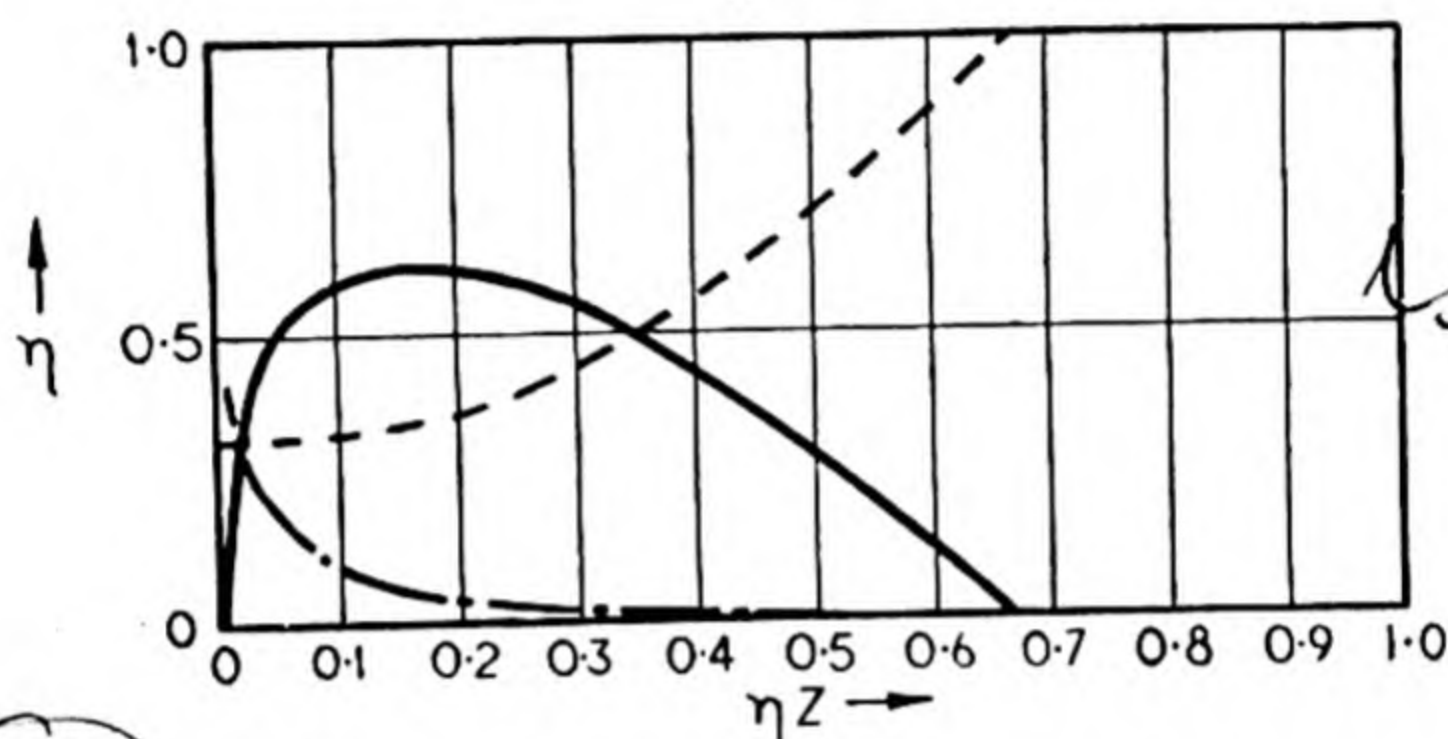
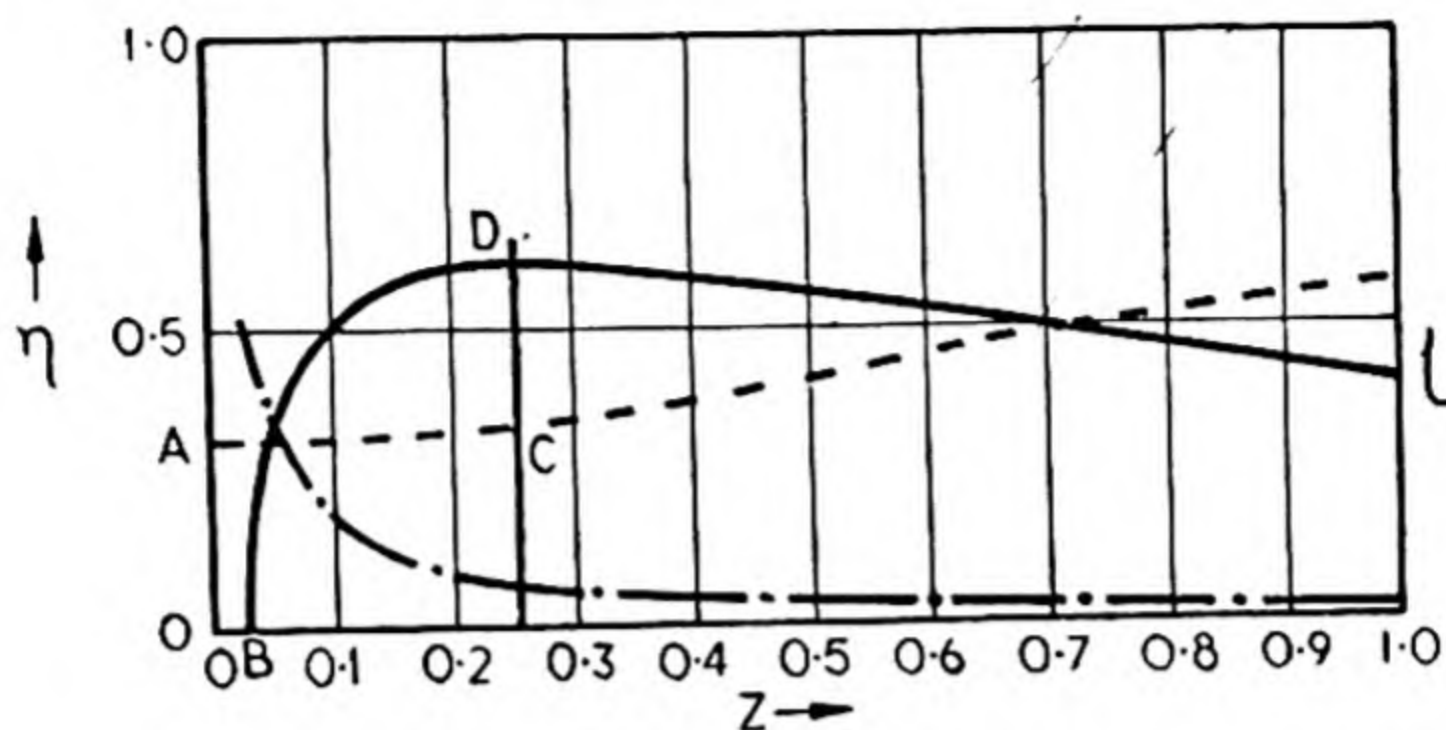


Fig. (7) **a** and **b**. Efficiency Curves for a Typical Heating System.

(a) Top: Efficiency versus fuel input  $Z = \frac{v}{\alpha_u L \beta}$

(b) Bottom: Efficiency versus useful heat  $\frac{H}{\alpha_u L \beta t_o}$

Key:

- Efficiency =  $H_u/H_a$ ;
- Fractional flue loss =  $H_b/H_a$ ;
- . - . - . Fractional wall loss =  $H_w/H_a$

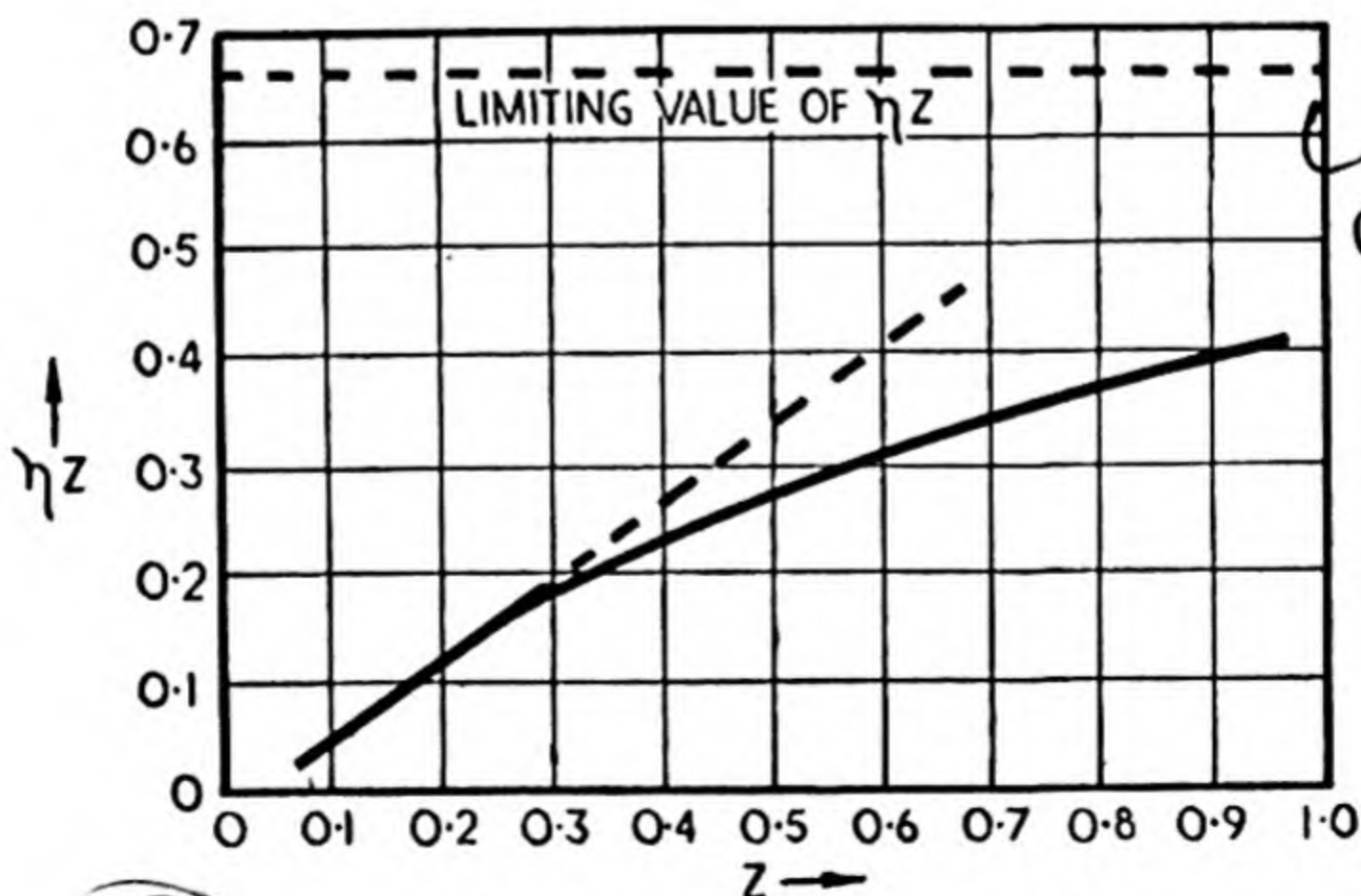


Fig. (7) **c**. Relation of Useful Heat ( $\eta Z$ ) to Heat Input.



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then falls as the "heat input variable"  $z$  (see (iii) below for definition) is continuously increased.

$$(iii) \quad z = \frac{c_r \cdot G' \cdot c_p}{\alpha_u \cdot \beta}$$

which may be called the "heat input variable" since it expresses the effect upon the efficiency of varying the heat input

where  $c_r$  = fuel burnt in lb./hr. ft.<sup>2</sup> of useful heat surface ;

$G'$  = weight of combustion products in lb./lb. of fuel ;

$c_p$  = mass specific heat of combustion products (C.H.U./lb. ° C.) ;

$\alpha_u$  = effective mean coefficient of heat transfer between combustion gases and charge (C.H.U./ft.<sup>2</sup> hr. ° C.) ; and

$\beta$  = the lowering of the theoretical flame temperature expressed as a dimensionless factor to represent the effect of delayed combustion.

The analysis thus shows that  $z$  is the correct dimensionless criterion containing the heat input. In plotting experimental results on a combustion appliance  $\alpha_u$  and  $\beta$  will not in general be known, but this calculation does indicate that the most fundamental quantity in which to express the fuel input is  $c_r \cdot G' \cdot c_p$ , i.e. the heat capacity of the total combustion products produced per unit of heat surface area and per hour (C.H.U./ft.<sup>2</sup>/hr. ° C.—the same dimensions as a heat transfer coefficient, as it should be, since  $\beta$  is a pure ratio). This shows, for example, that the well-known formula of Orrok<sup>7.27</sup> for boilers is not based on a sound dimensional analysis, since it involves  $G' \sqrt{(c_r)}$  instead of  $G' \cdot c_r$  as the "heat input variable."\*

The dimensionless quantities  $a$  and  $b$  are parameters which govern the shape of the curves obtained by plotting the efficiency  $\eta = H_b/H_a$  and the relative wall heat loss ratio  $H_w/H_a$  against  $z$  defined as follows :

$$(iv) \quad \text{The quantity } b \text{ is defined by the equation } b = \frac{t_u}{\beta t_a}$$

where  $t_u$  = temperature of charge above atmosphere, ° C. ;

$t_a$  = temperature obtainable by complete combustion with the given excess air but without heat loss ; and

\* If  $\alpha_u$  (the heat transfer coefficient to the charge),  $\beta$  (the combustion delay factor) and  $c_p$  are constant, as they may often be assumed to be, the product  $G'c_r$  is the only variable part of  $z$ .

<sup>7.27</sup> ORROK: *Mech. Eng.*, **48**, 3, p. 218. *A.S.M.E. Trans.*, **47**, p. 1184 (1925).



$b$  expresses the availability of the heat of combustion for the purpose in hand, namely, for supplying heat at a temperature  $t_u$ , since only the fraction  $1 - t_u/\beta t_a$  of the total heat of combustion is available as heat at  $t_u$ .  $b$  can therefore be called the "heat availability parameter," since it is a constant for any one heating system\* when the heat input varies. Calculation shows that  $b$  varies between  $\frac{2}{3}$  in the case of a high-temperature furnace, and about  $\frac{1}{10}$  in the case of some shell-type low-pressure boilers. Great importance attaches to the manner in which  $b$  varies with (1) the theoretical flame temperature (and hence the excess air), with (2) the slowness of combustion as expressed in the factor  $\beta$ , and (3) with the working temperature  $t_u$ .

(v) The quantity  $a$  is defined by the equation  $a = \frac{\alpha_w A_w}{\alpha_u A_u}$

where  $\alpha_w$  = effective mean heat transfer coefficient through the wall expressed in terms of  $t_u$  C.H.U./ft.<sup>2</sup> hr. ° C. ;

$A_w$  = wall area in channel for heat losses ft.<sup>2</sup> ; and

$A_u$  = heating surface area of charge, ft.<sup>2</sup>

$a$  is the dimensionless parameter which takes account of the wall heat losses, and can be called the "wall loss parameter." It is smallest in high-temperature furnaces, where the radiation heat transfer between the gases and the charge is very good and the walls are well insulated—calculation shows that here values of the order of 0.03 to 0.01 are usual. For Lancashire boilers the value of  $a$  may be as high as 0.1 if the walls of the brick setting are thin and of large area in relation to the useful heat absorbing area.

The exponential term  $e^{-1/z}$  in equation 7/8 is the term which results in the efficiency falling off and the flue gas losses rising when  $z$  exceeds a certain optimum value. This term is due to the heat transfer bottleneck between the heating gases and the heated charge ; as the heat input rises there is a slight increase in the mean temperature difference between the gas and the charge, but this is more than offset by the fact that more heat has to be transferred to obtain an equal efficiency. For a very large heat input the gases will not be appreciably cooled by the heating surfaces and hence all their energy will become flue losses.

The fact that  $z$  is the ratio of the heat capacity of the total combustion gases to the product of the heat transfer coefficient and the

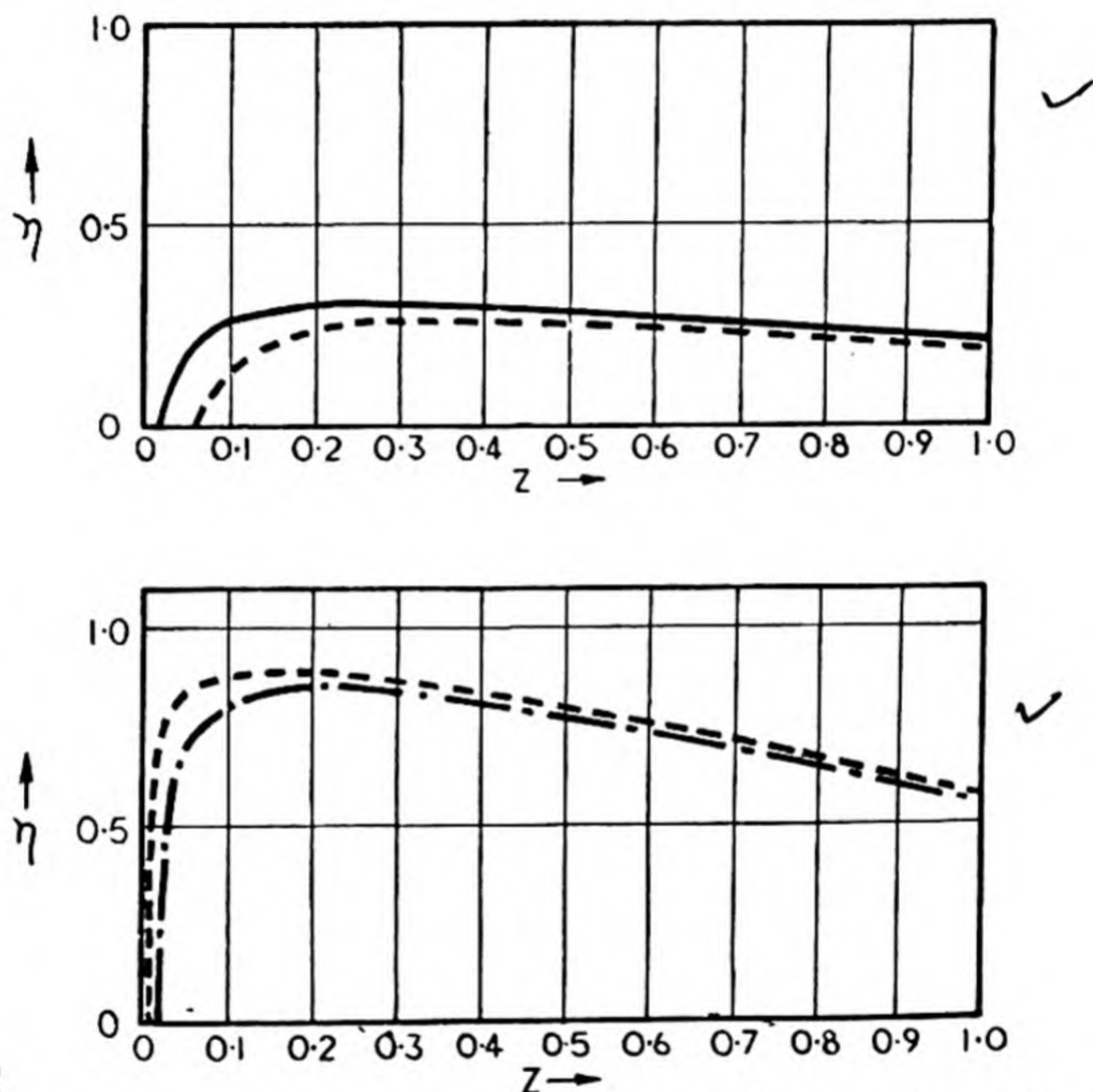
\* So long as the rapidity of combustion ( $\beta$ ) and the theoretical temperature of combustion ( $t_a$ ) are kept constant. It will of course be altered by variations in the excess air, since this alters  $t_a$ .



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delayed combustion factor  $\beta$  shows that this factor  $\beta$  is equivalent to a reduction in the heat transfer coefficient as well as to a reduction in the heat availability. The importance of rapid combustion for a high fuel economy is thus twofold: it is necessary in order to give as much of the heat in a usable form as possible and in order to give as rapid a use of this heat as possible.

These points are brought out in Figs. (7)11 and (7)12, which show



3.8

Fig. (7)12a and b. The Effect of Varying Heat Availability and of Varying Heat-Transfer Ratio on the Efficiency Curves.

(a) Top: Heat availability factor  $\frac{t_u}{\beta t_a} = \frac{2}{3}$

(b) Bottom: Heat availability factor  $\frac{1}{10}$

Key:

————  $\frac{A_w a_w}{A_u a_u} = 0.011;$

-----  $\frac{A_w a_w}{A_u a_u} = 0.003;$

- . - . -  $\frac{A_w a_w}{A_u a_u} = 0.10.$



the calculated curves of equations 7/8, 9, 10 against  $z$  (the dimensionless heat input variable) and against  $\eta z$  (the dimensionless useful output variable) for a particular choice of  $a$  and  $b$  (Fig. (7)11) and the variation in shape of the curve for  $\eta$  plotted against  $z$  when  $a$  and  $b$  are altered (Fig. (7)12). In Fig. (7)11a the three fractions into which the heat input is divided  $\left(\eta, \frac{H_w}{H_a}, \frac{H_b}{H_a}\right)$  are shown as functions of  $z$  and  $\eta z$ . It will be seen that  $\eta$  is zero up to a certain value of  $z$  (the point  $B$  on Fig. (7)11a), since smaller heat inputs than this are not sufficient to make up the wall losses and so bring the channel up to the working temperature. In the region  $BD$   $\eta$  rises steadily, and this rise is almost entirely due to the falling value of  $H_w/H_a$  (the wall losses  $H_w$  are constant for the given assumptions and hence  $H_w/H_a$  diminishes according to a rectangular hyperbola). In this region the flue losses are practically constant, the gases leaving very nearly with the heat corresponding to zero output. Beyond  $D$  the efficiency falls off again because of the rise in flue losses due to difficulty in conveying the heat from the gases to the charge with the limited heating areas available. At very high heat inputs practically all the heat leaves in the flue gases, so that  $\eta$  falls off asymptotically to zero as  $z$  tends to infinity. This is clear from the equations set out above, for the greatest amount of heat that can be transferred to the charge is evidently given by

$$H_u = \alpha_u \cdot A_u (\beta t_a - t_u) = (1 - b) \alpha_u \cdot A_u \cdot \beta \cdot t_a \quad (7/11)$$

which corresponds to the condition in which the effective temperature of the gas is  $\beta t_a$  throughout the system; and hence, for large values of  $z$ ,  $\eta = \frac{H_u}{H_a} = \frac{1 - b}{z}$ .

In Fig. (7)11b the same three fractions are plotted as functions of the useful heat expressed in a dimensionless way (i.e. the product  $\eta z$  of the efficiency  $\eta$  and the "heat input variable"  $z$ :  $\eta z = \frac{H_u}{\alpha_u A_u \beta t_a}$ ). Here the curves end at the point  $\eta z = 1 - b$ , since this is the greatest obtainable value for the useful heat, but they start at  $\eta z = 0$ .

Fig. (7)11c, in which the useful heat  $\eta z$  is plotted against the "heat input variable"  $z$ , shows that the useful heat is zero up to the value of  $z$  corresponding to a heat input sufficient to bring the system up to the working temperature  $t_u$ , and then rises almost linearly up to values of  $z$  which correspond (as can be seen from Fig. (7)11a) to the peak efficiency. The physical explanation of



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this is that the resistance to heat transfer to the charge is not important in this region, so that the flue losses are independent of  $z$ . Hence in this region the exponential terms in the formula can be neglected and it can be shown that in this case the useful heat  $H_u$  is related to the heat input  $H_a$  by the relation

$$H_a = \frac{H_u}{\gamma} + H_{ao} \quad . \quad . \quad . \quad . \quad . \quad (7/12)$$

where  $\gamma = 1 - b(1 - a)$ , i.e.  $\gamma$  is a fraction less than 1 and  $H_{ao} = H_w/\gamma$ , which is the heat input necessary to keep the system at the working temperature with no output.

Equation 7/12 may be written  $\gamma(H_a - H_{ao}) = H_u$ ; in this form it shows that  $\gamma$  is the fraction of the heat input additional to  $H_{ao}$  which is available for heating the charge at temperature  $t_u$ . The remainder of this additional input  $(1 - \gamma)(H_a - H_{ao})$ , is the additional flue loss due to the increased volume of combustion gas. This is an illustration of the principle enunciated in Chapter 2, section 2.2.7, that the higher the working temperature the greater is the amount of energy which must be liberated by combustion to obtain a given amount of heat at this working temperature. Equation 7/12 applies only to heating systems operating in the region where the flue temperature is practically constant for variations in  $z$  or  $\eta z$ . A linear equation of this type is found to be applicable to most industrial furnaces. Fig. (7)12a shows the curves for  $\eta$  expressed as a function of  $z$  calculated for a heat availability factor  $b$  of 0.67 for wall loss factors  $\alpha$  of 0.011 and 0.033 corresponding to high-temperature furnaces with good and bad insulation respectively. The peak efficiency is rather below 0.33, being lower for the less insulated furnaces. The poorly insulated furnace evidently does not begin to give useful heat until the input is nearly three times as great as for the well insulated one (i.e.  $H_{ao}$  in equation 7/12 is three times as large), and even then the useful heat rises more slowly. Fig. (7)12b relates to an appliance producing very low-grade heat, e.g. steam at 100° C. In this case the heat availability factor is 0.10, i.e. 90% of the heat of combustion could, if there were no wall losses, be supplied to the steam. Here the increase of the wall loss factor causes an appreciable drop in the efficiency at low ratings, but very little at high ratings. The wall loss factor is the same (0.033) for the lower curve of Fig. (7)12a as for the upper one of Fig. (7)12b, and so the two curves may be directly compared to show the effect of changes in the heat availability factor.



These formulæ have been compared with results obtained on glass furnaces (formulæ quoted in Section 7.3.3) and with formulæ obtained on a Lancashire boiler, from which it has been shown that furnaces usually operate in the part of the curve where the efficiency rises as the heat input is increased, while the Lancashire boiler owing to the limitation of heat transfer surface is operated in a region where the efficiency falls as the heat input is reduced.

These formulæ can be made use of in a number of ways, as follows:

(a) *Use of the results to decide what factors are most likely to increase the efficiency of a system.* It has been shown that if the efficiency of a heating system rises with increased output it is operating in such a way that the wall losses are as significant as the flue losses in determining the efficiency. Many furnaces are found experimentally to be in this category. If, on the other hand, the efficiency falls with increased output (as has been shown experimentally to be the case in boiler firing) the wall losses are not very significant, so that increasing the insulation cannot be expected to improve the efficiency by more than 2% or 3% at the most. In this case the variation of efficiency with output is almost entirely due to the bottle-neck for heat transfer between the gases and the charge. Hence marked improvement in fuel consumption in this case can only be expected from reduction of the flue losses which can be attained by improvement in the burner factor  $\beta$  (more rapid combustion), in the heat transfer (by increase of  $\alpha_u$  or  $A_u$ ) or by increase in the theoretical flame temperature (reduction of excess air; increase of  $\text{CO}_2$  content of flue gases). Thus it is possible by determining experimentally the variation of efficiency with fuel input or product output to deduce the gain which may be expected by improved insulation or improved combustion efficiency.

One of the most interesting consequences of these calculations is that improvement in the rapidity of combustion is just as likely to produce improvements in efficiency as is improvement in the  $\text{CO}_2$  content of the flue gases. This can be seen from the example of the Lancashire boilers discussed in the paper referred to<sup>7.20</sup>: an increase of  $\beta$  from 0.51 to 1.00 would have nearly halved the quantity  $b$  and so have raised the peak efficiency by about 8%; and it would have nearly doubled the value of  $c$ , (the heat input per unit of heating surface) at which the peak efficiency occurs.

Another interesting consequence of the general picture of a heating system put forward in this paper<sup>7.20</sup> is the possibility in



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certain cases of designing a plant so that the average heat input will occur at about the value corresponding to the peak efficiency. Where this is not practicable, for example because this would imply too large a heating surface and hence too high a capital cost, the desirability of installing waste-heat equipment to take account of the necessarily enlarged flue losses is clear.

(b) *Use of the formulæ to calculate the probable performance curves of a system.* Where the coefficients of heat transfer and of heat capacity are known, the formulæ (7/8, 7/9 and 7/10) can be used to calculate the expected variation of efficiency, flue loss, and wall loss with rating. They correspond to a somewhat simplified system, but the general way in which the efficiency at first rises, reaches a maximum and then falls as the heat input is concerned, will certainly be correct for all heating systems. It would be desirable when a new appliance is being designed to calculate the probable efficiency for heat inputs somewhat above and below the normal figure and so to ascertain on what portion of the general curve the appliance will be operating.

(c) *Use of the formulæ to deduce the heat transfer and combustion factors when the efficiency-output curves have been determined experimentally.* If the performance curves have been determined experimentally over a sufficiently wide\* range it is possible to use them together with the formulæ (7/8, 7/9 and 7/10) to deduce the appropriate parameters which govern the heating system. Three cases arise:

(1) *Where the efficiency rises approximately linearly with increased fuel input (furnace heating).* In this case the flue loss fraction can be treated as constant and the values of  $\gamma$  and  $H_{ao}$  can be calculated by fitting the linear equation (7/12) to the points. At any loading in this range the wall losses are then given as a fraction of the energy input by  $\gamma \cdot \frac{H_{ao}}{H_2}$  and the flue loss fraction by  $1 - \gamma$ .

(2) *Where the experimental values include the peak efficiency.* At this point it may be shown that the flue loss ratio  $H_f/H_a$  is very nearly equal to  $t_u/\beta t_a$ . Hence if the flue loss is known at this point  $t_u/\beta t_a$  can be calculated, and since  $t_u$  and  $t_a$  can be calculated  $\beta$  can be directly determined. Thus a measure of the combustion factor is obtained.

\* It should be stressed that these curves must correspond to constant combustion conditions so that if, for example, the  $\text{CO}_2$  varies with the output a correction must be made to make all the points equivalent.



(3) *Where the efficiency falls with increasing output over the whole range of points.* In this case determination of the efficiency alone can be used to determine the relative importance of flue and wall losses provided the experimental values cover a sufficiently wide range of  $z$ ; the method is to find which curve of the family shown in Fig. (7)12 fits the results best. The values of the parameters  $\frac{\alpha_w A_w}{\alpha_u A_u}$  and  $\frac{t_u}{\beta t_a}$  corresponding to this curve are thus obtained and the curves for wall losses and flue losses can be deduced from equations (7/9) and (7/10).

## 7.5. Inventing a New Furnace

For completeness it is necessary to discuss the case where the aim is the most radical one possible, namely, that of testing and developing apparatus based on new ideas. Naturally, no general recipe can be given for invention, but nevertheless it is possible to observe certain characteristic features which have been present in the history of all successful inventions. The process can usefully be discussed using the invention of the regenerative furnace by Sir William Siemens as a typical illustration. Siemens found the steel industry using crucibles in coke beds and puddling furnaces, and left them with the open-hearth furnace almost in its modern form.

The first essential of a successful invention, which was satisfied in the case of Siemens, is that the *need* for the new process must be "in the air." That this was so is shown by the fact that another process was being developed at almost the same time for making cheap steel, which has only been overshadowed by the open-hearth furnace in the last generation. This was the converter process invented and developed by Sir Henry Bessemer. Clearly, the economic position must be ripe for a new invention if this invention is to achieve marked success.

The second essential is the origination of the new idea or new principle on which the new process is to be based.

It often happens that an idea is already available but has not yet been brought to a form where it can be used. In the case of Siemens and the open-hearth furnace the new idea had not previously been applied to furnaces, but it is on record that it was suggested to Siemens by a patent taken out some years before for an apparatus to assist tubercular patients. This apparatus consisted of a chamber with metal fins inside, through which the breath



of the patient passed in and out. Outgoing breath warmed the fins so that the incoming air had the chill taken off it. The thought process connecting this apparatus with the regenerator of a modern furnace is fairly clear, the only essential addition being the fact that the operation of the furnace must be made intermittent like the breath of the patient.

Having achieved his new idea, the next and most heartbreaking series of steps involved the overcoming of material difficulties, i.e. the construction of furnaces which would not melt or otherwise collapse when heated by the regenerative system. This step necessarily involved a considerable expenditure of money, which was only possible in Siemens' case because he had already amassed a large private fortune, mainly through the successful sale of a water meter. Certain principles for achieving success in this step with a minimum expenditure can, however, be laid down. The study of whether a new idea is at all practicable is best done first on a relatively small scale—the pilot plant stage of the chemical industry. For example, the general shape of a furnace system and the decision whether to fire it with a separate gas-producer or with built-in coal fires can be studied on a very small scale so that the whole apparatus can continually be pulled down and re-erected with a quite new shape. On the other hand, the thermal efficiency and questions of whether materials will stand up to the working conditions must be studied on a scale comparable with the final practicable one. Even here, however, the experimental furnaces should be designed in such a way that they can readily be rebuilt and altered in shape, for example, removable crowns and separate side walls may be desirable even where they would not be economic in the final furnace. The quenching method discussed in 7.2 is also of value in these tests. Nowadays it is rarely possible to solve these problems by the means provided by a personal fortune, and consequently they must be done either by central research bodies or by the departments of industrial firms. In either case this may tend to make greater difficulties even than those which Siemens had, as it is liable to bring in the last stage too soon.

This last stage is one in which even more inventions fail than in that of overcoming the resistance of the apparatus itself. This is the demonstration that the new is more economic than the old. Siemens had great difficulty at this stage with his new process and it was not until he built his own experimental steel works that the adoption of his methods was seriously considered. Today, when



the capital invested in plant is very much greater than that in Siemens' time and when existing plant is already adequate to supply the world's demands, so that the new processes have only the advantage of possible reductions of fuel consumption or improvement of quality of product, the resistances in this final stage are liable to be greater than they were in the time of Siemens. The ideal place for new developments is a country which is just starting to build the industry from nothing. Nevertheless it is believed that an era in which the furnace becomes an instrument of precision, instead of the Cinderella of the factory, is dawning, and it is certain that advances will be made both in the direction of increasing the knowledge discussed in this monograph and in that of inventing new processes using wherever possible this existing knowledge.

## SUMMARY OF CHAPTER 7. THE APPLICATION OF THE SCIENTIFIC METHOD TO FURNACES

### 7.1. The Limitations of the Scientific Method

The scientific method is ideally only applicable to exactly specified, isolated systems. In the furnace neither of these conditions is even approximately satisfied. This chapter deals with the consequences of this fact, and the methods for obtaining benefit from scientific results in spite of it. These methods are different according to the four objectives of study.

### 7.2. Diagnosis of a Single Furnace

**7.2.1. The Aim of Diagnosis.** The methods of diagnosis are applied when it is desired to improve the working of an individual furnace which is not fulfilling expectations.

**7.2.2. Probes.** Probes are used to see what is going on inside a furnace. Probes may be designed for gas analysis, for measurements of temperature, heat-flow, aerodynamic pressure and velocity, and for sampling solid and liquid products.

**7.2.3. The Diagnosis of Furnace Faults from Probe Measurements.** (a) The causes of irregularity of product can be deduced from thermal measurements and gas analysis in the heating chamber. (b) The causes of excessive fuel consumption can be deduced by setting up a heat balance. (c) The causes of excessive wear of refractories can be traced by detailed gas analysis and gas-flow probing or by a heat-flow meter.



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**7.2.4. The Quenching Method.** An alternative method to probing the hot system is to cool it down, under conditions that will not disturb the factor of interest, and then open it up.

**7.2.5. Normal Furnace Instrumentation.** The installation of complete instrumentation schemes for combustion control, process control and safeguarding the furnace is becoming an essential on all large furnace systems.

## 7.3. Statistical Study of a Furnace Type

**7.3.1. The Scope of the Statistical Method.** Statistics may be used either as a method of comparing the performance of furnaces of the same type but which differ both as regards design factors and output factors or to evaluate the effect of control variables upon the performance.

**7.3.2. Specification of the Variables.** These must be chosen from experience as to what factors are likely to be important and which of these are under the control of the designer and operator.

**7.3.3. Construction of the Formula.** The formula is constructed from actual measurements as far as possible, but in practice has to be supplemented to some extent by theoretical assumptions. Statistical methods of constructing the formula from observations are given.

**7.3.4. Use of Formulæ to Compare Different Furnaces.** If the method is correctly applied the final result is to enable the user to decide which design factors are important and what is the optimum value of them.

## 7.4. Designing a Modification of an Existing Type

**7.4.1. The Use of Formulæ.** The essential step in applying science to furnace design is the right choice of assumptions to make the actual reality sufficiently simple to be handled.

**7.4.2. Model Tests.** Models provide a method of solving cases which are too complicated for a completely theoretical treatment.

**7.4.3. General Laws Governing a Combustion-heated Furnace.** A general set of equations are developed which show that the thermal efficiency first rises to a maximum and then falls as the output of a given furnace is steadily increased from zero. From the shape of the experimental curve of efficiency against output it is possible to deduce which is the most profitable method of increasing the efficiency.



### 7.5. Inventing a New Furnace

The steps in the history of a successful invention are (1) that the need must be "in the air." (2) That the new principle must be discovered or available. (3) The overcoming of material difficulties by a series of experimental units of increasing size. (4) Demonstrating that the new process is economic. It is believed that by some such steps as those described in this chapter the furnace will gradually become an instrument of precision.



# TABLE OF CONVERSION FACTORS

Mass: 1 lb.=453.6 gm. 1 gm.=0.002205 lb. 1 metric tonne=0.9842 British tons (2240 lb.)  
 1 British ton=1.016 tonnes.

Length: 1 foot=30.48 cm. 1 cm.=0.03281 ft.

Area: 1 ft.<sup>2</sup>=929.0 cm.<sup>2</sup> 1 cm.<sup>2</sup>=0.001076 ft.<sup>2</sup>

Volume: 1 ft.<sup>3</sup>=28320 cm.<sup>3</sup> = 28.32 litre.

1 litre=0.03531 ft.<sup>3</sup>

Imperial gallon=4546 cm.<sup>3</sup>=4.546 litre.

Temperature:  $T_C = \frac{T_F - 32}{1.8}$  if  $T_C$ =temperature "Celsius" (Centigrade)  $T_F$ =temperature Fahrenheit.

$T_F = 1.8 T_C + 32$

Absolute zero = -459.8 ° F.

= -273.2 ° C.

Acceleration due to Gravity:  $g = 981.3 \text{ cm./sec.}^2$ ;  $32.2 \text{ ft./sec.}^2$

Density: 1 gm./c.c.=62.43 lb./ft.<sup>3</sup>; 1 lb./ft.<sup>3</sup>=0.01602 gm./c.c.

Pressure: 1 atmosphere=760 mm. Hg (29.92 in. Hg)

=10.33 m. H<sub>2</sub>O (33.90 ft. H<sub>2</sub>O)

=1.033 kg./cm.<sup>2</sup>=14.70 lb./in.<sup>2</sup>

1 mm. H<sub>2</sub>O=0.0394 in. H<sub>2</sub>O=0.000100 kg./cm.<sup>2</sup>=0.001422 lb./in.<sup>2</sup>

Energy: 1 C.H.U.=0.4536 k.cal.=1.8 B.Th.U.=1400 ft./lb.=1899 joule (watt-second)

1 kW.=0.5266 C.H.U./sec.

1 kWH.=1896 C.H.U.

Thermal conductivity:  $\frac{1 \text{ C.H.U./in.}}{\text{ft.}^2 \cdot \text{hr.} \cdot ^\circ \text{C.}} = \frac{1 \text{ B.Th.U./in.}}{\text{ft.}^2 \cdot \text{hr.} \cdot ^\circ \text{F.}} = 0.0003445 \frac{\text{cal./cm.}}{\text{cm.}^2 \text{ sec.} \cdot ^\circ \text{C.}} = 0.08333 \frac{\text{C.H.U./ft.}}{\text{ft.}^2 \cdot \text{hr.} \cdot ^\circ \text{C.}}$



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